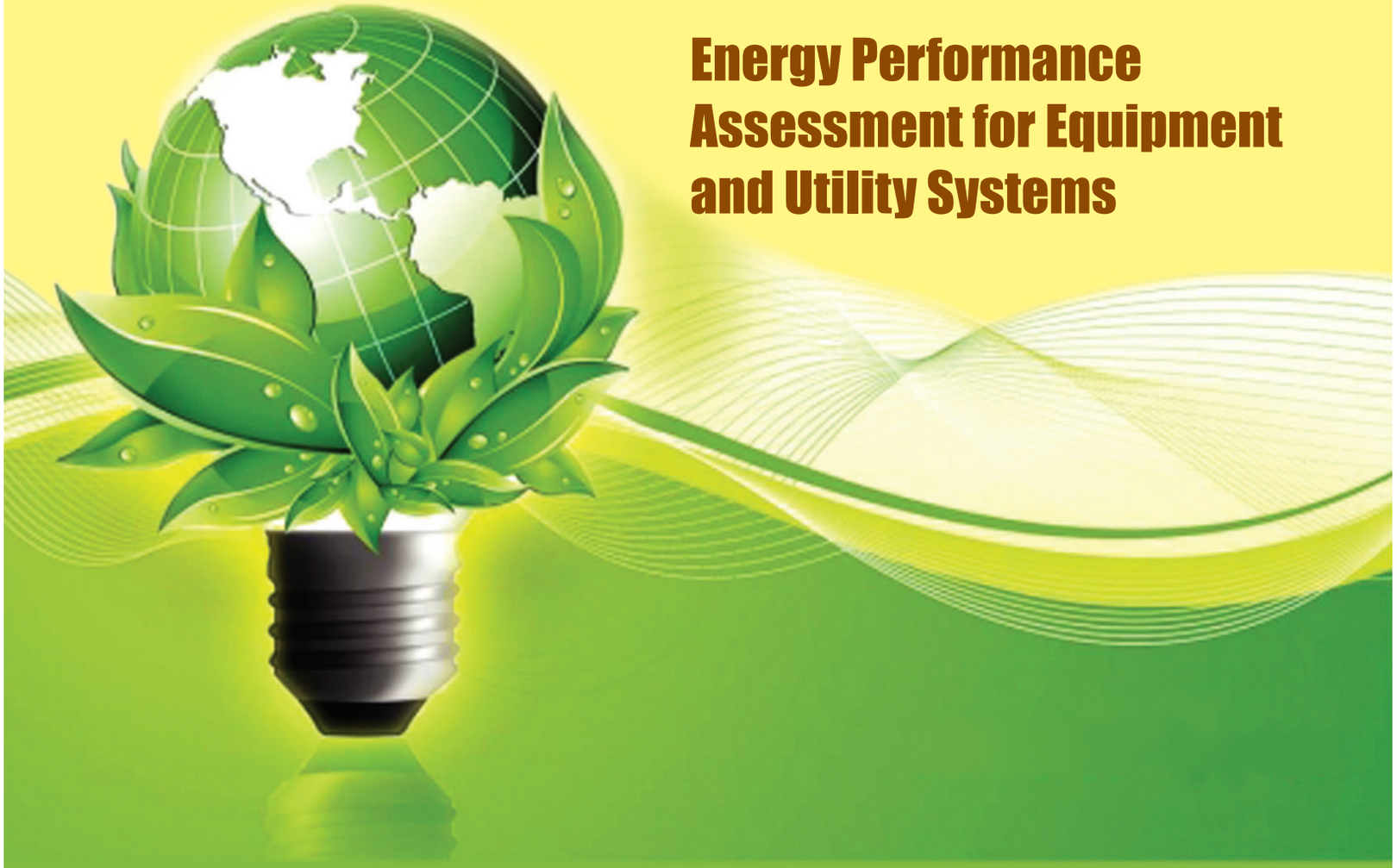
4

Energy Performance Assessment for Equipment and Utility Systems



GUIDE BOOK

for

NATIONAL CERTIFICATION EXAMINATION

FOR ENERGY MANAGERS AND ENERGY AUDITORS

Bureau of Energy Efficiency

(A Statutory Body under Ministry of Power, Government of India)

4th Floor, Sewa Bhawan, R. K.Puram, New Delhi - 110 066

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FOURTH EDITION, 2015

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PREFACE

National Certification Examination for Energy Managers and Energy Auditors aims to develop qualified professionals in the field of energy conservation in India.

At the initiative of Bureau of Energy Efficiency (BEE) for meeting the requirements of National Certification Examination for Energy Managers and Energy Auditors, Dr.Ambedkar Institute of Productivity, Chennai, the training institute of National Productivity Council of India has prepared four guide books in respect of four examinations as follows:

Guide Book	Examination Paper	Topic
Book-1	Paper-1	General Aspects of Energy Management and Energy Audit
Book-2	Paper-2	Energy Efficiency in Thermal Utilities
Book-3	Paper-3	Energy Efficiency in Electrical Utilities
Book-4	**Paper-4**	**Energy Performance Assessment for Equipment and Utility Systems**

This book – 4 on Energy Performance Assessment for Equipment and Utility Systems focuses on practical methods of assessing the performance in the field. Performance assessment requires the application of variety of instruments and numerous ways by which different parameters could be assessed. This book attempts to focus on these aspects, which would enable an energy auditor to carry out systematic and comprehensive performance assessment in order to identify areas for improvement.

These books have been prepared with material support from the various books referenced at the end of each chapter. We trust that these books, besides its intended purpose, will also help in enhancing the knowledge base of practicing engineers and managers in understanding and applying energy efficiency techniques.

In the areas where there is a lack of information, the contents should not be treated as standards or absolute requirements, but serve as a base for further improvements. Hence we invite your valued suggestions and comments, which may be mailed to **guidebooks@em-ea.org.**

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March 17, 2015

MESSAGE

India's economic growth depends on the long-term availability of energy from sources that are affordable, accessible and environmentally friendly. Increasing demands for fossil fuels have led to over exploitation of natural resources with adverse impacts on environment and the resultant health issues.

Energy efficiency and its conservation is one of the most cost effective measures to bridge the gap between demand and supply, and to mitigate the environmental impacts of rapid development.

It is heartening to note that the guide books prepared for the National Certification Examination for energy managers and auditors has caught the attention of a wide cross section of the engineering fraternity, not only in India but also in many countries around the world. Many of the techniques and technologies enumerated in these guide books towards enhancing energy efficiency have been put to practice and rich dividends have been realized. Thus these books besides meeting the requirements of the examination are also serving as a manual for practicing engineers.

I commend the efforts put in to prepare such a practical and comprehensive compilation. My best wishes to the users of these books.

(Pradeep K.Sinha)

ENERGY IS LIFE
B E E
C NSERVE IT

ऊर्जा दक्षता ब्यूरो
(भारत सरकार, विद्युत मंत्रालय)

BUREAU OF ENERGY EFFICIENCY
(Government of India, Ministry of Power)

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Director General

MESSAGE

The guide books for National Certification Examination for Energy Managers and Energy Auditors are an enthusiastic celebration of a wide variety of topics related to energy efficiency, especially those which are of practical nature, intricate structure and timely relevance. A glance through the pages of this book will show that it is an unusual type of publication. Though primarily intended for the purpose of being a study material for the examination these guide books have been found to be an out and out a practical guide to implementing energy savings measures in industry and establishments. That they have also been adopted by the energy efficiency promotion organisations of other countries is a testimony to the efficacy of these guide books. Also many of the Indian universities have adopted these books for Post Graduate programmes in energy management.

Written by a group of experienced professionals practicing energy efficiency the books explore the depth and breadth of various possibilities for saving energy in a wide gamut of processes and equipment. Guides such as these are no replacement for the necessary hard work required to realize the potential of energy efficiency but they can certainly help you ensure that your efforts are focused and effective.

I wish all the readers the very best.

(Ajay Mathur)

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ACKNOWLEDGEMENT

Energy efficiency has assumed a critical role in our pursuit of sustainable development. The gains from energy efficiency have been harnessed the world over and India is no exception. Information from the industry and other establishments reveal the rapid strides made in the area of energy efficiency in our country. Examples of industries which have achieved this potential are projected in the energy conservation award programme of Government of India.

As the low hanging fruits appear to have been picked up the path to success in energy management is getting steeper and steeper. More and more information, innovations and knowledge are required to further exploit the energy conservation potential. The continuous efforts to upgrade and refine the guide books for National Certification Examination for Energy Managers and Energy Auditors are aimed at equipping the reader with the resources to identify and implement various energy conservation measures. Through the system of examination we have been able to build sufficient skilled manpower in the area of energy efficiency. We are pleasantly surprised by the enormous interest evinced in the examination from all corners

The unwavering support for the development of these guide books and constructive comments from various quarters about their contents, have been invaluable.

(Bhaskar J. Sarma)

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*** Note: The Chapters 16 and 17 are for reference and will not be a part of the examination syllabus until further notification.**

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Chapter 1

Energy Performance Assessment of Boilers

Learning Objectives

In this chapter you will learn about

- Boilers performance: terms and definitions
- Methods of boiler efficiency evaluation: direct and indirect
- Planning for efficiency evaluation
- How to evaluate efficiencies for oil and coal fired boiler?

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Conduct boiler efficiency test through direct and indirect method**
- ✓ **Perform efficiency calculations**

1. ENERGY PERFORMANCE ASSESSMENT OF BOILERS

1.1 Introduction

Performance of the boiler, like efficiency and evaporation ratio reduces with time, due to poor combustion, heat transfer fouling and poor operation and maintenance. Deterioration of fuel quality and water quality also leads to poor performance of boiler. Efficiency testing helps us to find out how far the boiler efficiency drifts away from the best efficiency. Any observed abnormal deviations could therefore be investigated to pinpoint the problem area for necessary corrective action. Hence it is necessary to find out the current level of efficiency for performance evaluation, which is a prerequisite for energy conservation action in industry.

1.2 Purpose of the Performance Test

- To find out the efficiency of the boiler
- To find out the evaporation ratio

The purpose of the performance test is to determine actual performance and efficiency of the boiler and compare it with design values or norms. It is an indicator for tracking day-to-day and season-to-season variations in boiler efficiency and energy efficiency improvements.

1.3 Performance Terms and Definitions

1. Boiler Efficiency, $\eta = \frac{Heat\ output}{Heat\ input} x 100$

$= \frac{\text{Heat in steam output, kcal}}{\text{Heat in fuel input, kcal}} \text{x } 100$

2. Evaporation Ratio $= \frac{Quantity\ of\ steam\ generation}{Quantity\ of\ fuel\ consumption}$

1.4 Scope

The procedure describes routine test for both oil fired and solid fuel fired boilers using coal, agro residues etc. Only those observations and measurements need to be made which can be readily applied and is necessary to attain the purpose of the test.

1.5 Reference Standards

British standards, BS845: 1987

The British Standard BS845: 1987 describes the methods and conditions under which a boiler should be tested to determine its efficiency. For the testing to be done, the boiler should be operated under steady load conditions (generally full load) for a period of one hour after which readings would be taken during the next hour of steady operation to enable the efficiency to be calculated.

The efficiency of a boiler is quoted as the % of useful heat available, expressed as a percentage of the total energy potentially available by burning the fuel. **This is expressed on the basis of gross calorific value (GCV).**

This deals with the complete heat balance and it has two parts:

- Part One deals with standard boilers, where the indirect method is specified
- Part Two deals with complex plant where there are many channels of heat flow. In this case, both the direct and indirect methods are applicable, in whole or in part.

ASME Standard: PTC-4-1 Power Test Code for Steam Generating Units

This consists of

- Part One: Direct method (also called as Input -output method)
- Part Two: Indirect method (also called as Heat loss method)

IS 8753: Indian Standard for Boiler Efficiency Testing

Most standards for computation of boiler efficiency, including IS 8753 and BS845 are designed for spot measurement of boiler efficiency. **Invariably, all these standards do not include blow down as a loss in the efficiency determination process**.

Basically Boiler efficiency can be tested by the following methods:

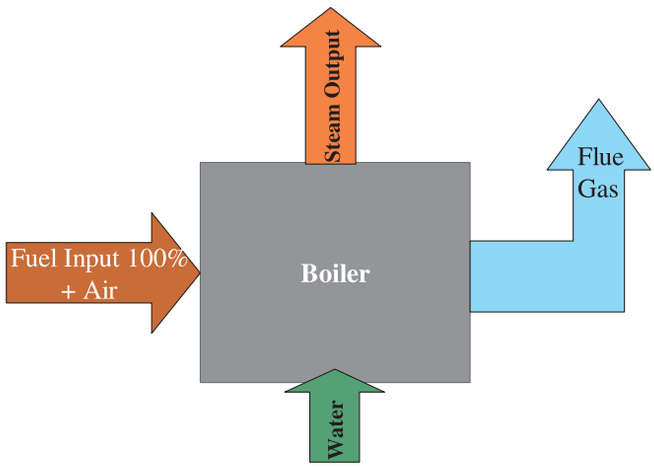
1) **The Direct Method:** Where the energy gain of the working fluid (water and steam) is compared with the energy content of the boiler fuel.
2) **The Indirect Method:** Where the efficiency is the difference between the losses and the energy input.

1.6 The Direct Method Testing

1.6.1 Description

This is also known as 'input-output method' due to the fact that it needs only the useful output (steam) and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula:

$$Boiler\ Efficiency = \frac{Heat\ Output}{Heat\ Input} x100$$



Boiler Efficiency

$$= \frac{\text{Steam flow rate, kg/hr} \times (\text{Steam enthalpy, kcal/kg} - \text{Feed water enthalpy, kcal/kg})}{\text{Fuel firing rate, kg/hr} \times \text{Gross calorific value, kcal/kg}} \times 100$$

1.6.2 Measurements Required for Direct Method Testing

Heat input

Both heat input and heat output must be measured. The measurement of heat input requires knowledge of the calorific value of the fuel and its flow rate in terms of mass or volume, according to the nature of the fuel.

For gaseous fuel: A gas meter of the approved type can be used and the measured volume should be corrected for temperature and pressure. A sample of gas can be collected for calorific value determination, but it is usually acceptable to use the calorific value declared by the gas suppliers.

For liquid fuel: Heavy fuel oil is very viscous, and this property varies sharply with temperature. The meter, which is usually installed on the combustion appliance, should be regarded as a rough indicator only and, for test purposes, a meter calibrated for the particular oil is to be used and over a realistic range of temperature should be installed. Even better is the use of an accurately calibrated day tank.

For solid fuel: The accurate measurement of the flow of coal or other solid fuel is very difficult. The measurement must be based on mass, which means that bulky apparatus must be set up on the boiler-house floor. Samples must be taken and bagged throughout the test, the bags sealed and sent to a laboratory for analysis and calorific value determination. In some more recent boiler houses, the problem has been alleviated by mounting the hoppers over the boilers on calibrated load cells, but these are yet uncommon.

Heat output

There are several methods, which can be used for measuring heat output. With steam boilers, an installed steam meter can be used to measure flow rate, but this must be corrected for temperature and pressure. In earlier years, this approach was not favoured due to the change in accuracy of orifice or venturi meters with flow rate. It is now more viable with modern flow meters of the variable-orifice or vortex-shedding types.

The alternative with small boilers is to measure feed water, and this can be done by previously calibrating the feed tank and noting down the levels of water during the beginning and end of the trial. Care should be taken not to pump water during this period. Heat addition for conversion of feed water at inlet temperature to steam, is considered for heat output.

In case of boilers with intermittent blowdown, blowdown should be avoided during the trial period. In case of boilers with continuous blowdown, the heat loss due to blowdown should be calculated and added to the heat in steam.

1.6.3 Boiler Efficiency by Direct Method: Calculation and Example

Test Data and Calculation

Water consumption and coal consumption were measured in a coal-fired boiler at hourly intervals. Weighed quantities of coal were fed to the boiler during the trial period. Simultaneously water level difference was noted to calculate steam generation during the trial period. Blow down was avoided during the test. The measured data is given below.

Type of boiler: Coal fired Boiler

Heat output data

Quantity of steam generated (output)	: 8 TPH
Steam pressure / temperature	: 10 kg/cm^2 (g)/ 180 ^{0}C
Enthalpy of steam (dry & saturated) at 10 kg/cm^2(g) pressure	: 665 kcal/kg
Feed water temperature	: 85 ^{0}C
Enthalpy of feed water	: 85 kcal/kg

Heat input data

Quantity of coal consumed (Input)	: 1.6 TPH
GCV of coal	: 4000 kcal/kg

Calculation

$$Boiler\ efficiency\ (\eta) = \frac{Q\,x\,(H-h)}{q\,x\,GCV} x100$$

Where **Q** = Quantity of steam generated per hour (kg/hr)
q = Quantity of fuel used per hour (kg/hr)
GCV = Gross calorific value of the fuel (kcal/kg)
H = Enthalpy of steam (kcal/kg)
h = Enthalpy of feed water (kcal/kg)

$$\text{Boiler efficiency}(\eta) = \frac{8, \text{TPH x } 1000, \text{ kg / T x } (665 - 85), \text{kcal / kg}}{1.6, \text{TPH x } 1000, \text{kg / T x } 4000, \text{kcal / kg}} \text{ x } 100$$

$$= \mathbf{72.5\%}$$

Evaporation Ratio = 8 Tonne of steam / 1.6 Tonne of coal
= 5

1.6.4 Merits and Demerits of Direct Method

Merits

- Plant people can evaluate quickly the efficiency of boilers
- Requires few parameters for computation
- Needs few instruments for monitoring

Demerits

- Does not give clues to the operator as to why efficiency of system is lower
- Does not calculate various losses accountable for various efficiency levels
- Evaporation ratio and efficiency may mislead, if the steam is highly wet due to water carryover

1.7 The Indirect Method Testing

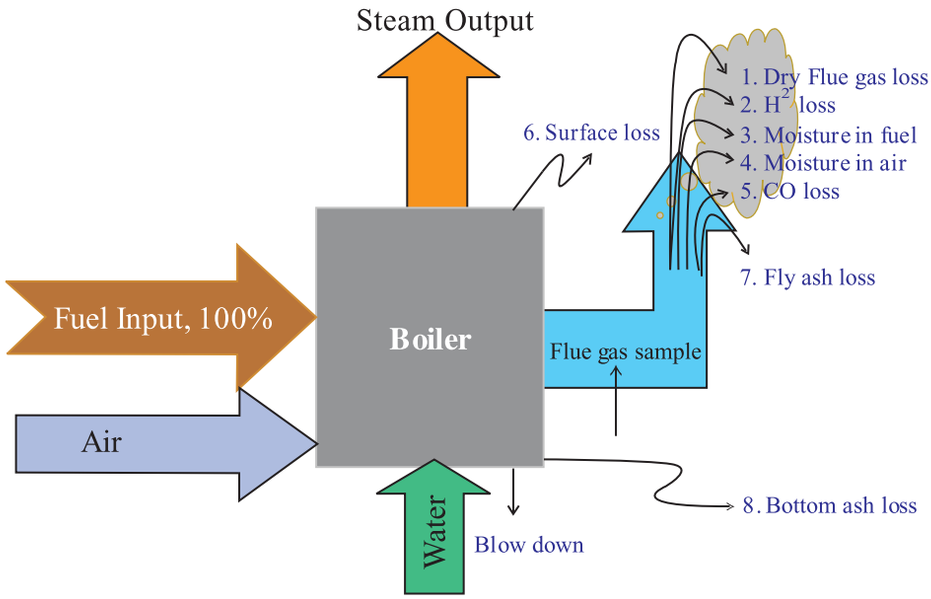
1.7.1 Description

The efficiency can be measured easily by measuring all the losses occurring in the boilers using the principles to be described. The disadvantages of the direct method can be overcome by this method, which calculates the various heat losses associated with boiler. The efficiency can be arrived at, by subtracting the heat loss fractions from 100.An important advantage of this method is that the errors in measurement do not make significant change in efficiency.

Thus if boiler efficiency is 90%, an error of 1% in direct method will result in significant change in efficiency. i.e.90 $\pm$ 0.9 = 89.1 to 90.9. In indirect method, 1% error in measurement of losses will result in

Efficiency = 100 – (10 $\pm$ 0.1) = 90 $\pm$ 0.1 = 89.9 to 90.1

The various heat losses occurring in the boiler are:



Efficiency = 100 - (1+2+3+4+5+6+7+8) (by Indirect Method)

The following losses are applicable to liquid, gas and solid fired boiler

L1- Loss due to dry flue gas (sensible heat)
L2- Loss due to hydrogen in fuel (H_2)
L3- Loss due to moisture in fuel (H_2O)
L4- Loss due to moisture in air (H_2O)
L5- Loss due to carbon monoxide (CO)
L6- Loss due to surface radiation, convection and other unaccounted*.

*Losses which are insignificant and are difficult to measure.

The following losses are applicable to solid fuel fired boiler in addition to above

L7- Unburnt losses in fly ash (Carbon)
L8- Unburnt losses in bottom ash (Carbon)

Boiler Efficiency by indirect method = 100 – (L1+L2+L3+L4+L5+L6+L7+L8)

1.7.2 Measurements Required for Performance Assessment Testing

The following parameters need to be measured, as applicable for the computation of boiler efficiency and performance.

a) Flue gas analysis

1. Percentage of CO_2 or O_2 in flue gas
2. Percentage of CO in flue gas
3. Temperature of flue gas

b) Flow meter measurements for

1. Fuel
2. Steam
3. Feed water
4. Condensate water
5. Combustion air

c) Temperature measurements for

1. Flue gas
2. Steam
3. Makeup water
4. Condensate return
5. Combustion air
6. Fuel
7. Boiler feed water

d) Pressure measurements for

1. Steam
2. Fuel
3. Combustion air, both primary and secondary
4. Draft

e) Water condition

1. Total dissolved solids (TDS)
2. pH
3. Blow down rate and quantity

The various parameters that were discussed above can be measured with the instruments that are given in Table 1.1.

Table 1.1 Typical Instruments used for Boiler Performance Assessment

Instrument	Type	Measurements
Flue gas analyzer	Portable or fixed	% CO_2 , O_2 and CO
Temperature indicator	Thermocouple, liquid in glass	Fuel temperature, flue gas temperature, combustion air temperature, boiler surface temperature, steam temperature
Draft gauge	Manometer, differential pressure	Amount of draft used or available
TDS meter	Conductivity	Boiler water TDS, feed water TDS, make-up water TDS.
Flow meter	As applicable	Steam flow, water flow, fuel flow, air flow

1.7.3 Test Conditions and Precautions for Indirect Method Testing

A) The efficiency test does not account for:

- ***Blow down loss.*** The amount of energy wasted by blow down varies over a wide range.
- ***Soot blower steam.*** The amount of steam used by soot blowers is variable that depends on the type of fuel.
- ***Auxiliary equipment energy consumption.*** The combustion efficiency test does not account for the energy usage by auxiliary equipments, such as burners, fans, and pumps.

B) Preparations and pre conditions for testing

- Burn the specified fuel(s) at the required rate.
- Do the tests while the boiler is under steady load. Avoid testing during warming up of boilers from a cold condition
- Obtain the charts / tables for the additional data.
- Determination of general method of operation
- Sampling and analysis of fuel and ash.
- Ensure the accuracy of fuel and ash analysis in the laboratory.
- Check the type of blow down and method of measurement
- Ensure proper operation of all instruments.
- Check for any air infiltration in the combustion zone.

C) Flue gas sampling location

It is suggested that the exit duct of the boiler be probed and traversed to find the location of the zone of maximum temperature. This is likely to coincide with the zone of maximum gas flow and is therefore a good sampling point for both temperature and gas analysis.

D) Options of flue gas analysis

Check the Oxygen Test with the Carbon Dioxide Test

If continuous-reading oxygen test equipment is installed in boiler plant, use oxygen reading. Occasionally use portable test equipment that checks for both oxygen and carbon dioxide. If the carbon dioxide test does not give the same results as the oxygen test, something is wrong. One (or both) of the tests could be erroneous, perhaps because of stale chemicals or drifting instrument calibration. Another possibility is that outside air is being picked up along with the flue gas. This occurs if the combustion gas area operates under negative pressure and there are leaks in the boiler casing.

Carbon Monoxide Test

The carbon monoxide content of flue gas is a good indicator of incomplete combustion with all types of fuels, as long as they contain carbon. Carbon monoxide in the flue gas is minimal with ordinary amounts of excess air, but it rises abruptly as soon as fuel combustion starts to be incomplete.

E) Planning for the testing

- The testing is to be conducted for a duration of 4 to 8 hours in a normal production day.
- Advanced planning is essential for the resource arrangement of manpower, fuel, water and instrument check etc and the same to be communicated to the boiler Supervisor and Production Department.
- Sufficient quantity of fuel stock and water storage required for the test duration should be arranged so that a test is not disrupted due to non-availability of fuel and water.
- Necessary sampling point and instruments are to be made available with working condition.
- Lab analysis should be carried out for fuel, flue gas and water in coordination with lab personnel.
- The steam table, psychometric chart, calculator are to be arranged for computation of boiler efficiency.

1.7.4 Boiler Efficiency by Indirect Method: Calculation Procedure and Formula

In order to calculate the boiler efficiency by indirect method, all the losses that occur in the boiler must be established. These losses are conveniently related to the amount of fuel burnt. In this way it is easy to compare the performance of various boilers with different ratings.

Conversion formula for proximate analysis to ultimate analysis			
	%C	=	0.97C+ 0.7(VM+0.1A) - M(0.6-0.01M)
	$\%H_2$	=	0.036C + 0.086 (VM -0.1xA) - $0.0035M^2$ (1-0.02M)
	$\%N_2$	=	2.10 -0.020 VM
where	C	=	% of fixed carbon
	A	=	% of ash
	VM	=	% of volatile matter
	M	=	% of moisture

However it is suggested to get a ultimate analysis of the fuel fired periodically from a reputed laboratory.

Theoretical (stoichiometric) air fuel ratio and excess air supplied are to be determined first for computing the boiler losses. The formula is given below for the same.

a) Theoretical air required for combustion	=	$[(11.6xC)+\{34.8x(H_2-O_2/8)\}+(4.35xS)]/100$ kg/kg of fuel. [from fuel analysis]
		Where C, H_2, O_2 and S are the percentage of carbon, hydrogen, oxygen and sulphur present in the fuel.
b) % Excess Air supplied (EA)	=	$\frac{O_2\%}{21-O_{2\%}}x100$ [from flue gas analysis]
		Normally O_2 measurement is recommended. If O_2 measurement is not available, use CO_2 measurement.

		$\frac{7900\,x[(CO_2\%)_t-(CO_2\%)_a]}{(CO_2)_a\%\ x\ [100-(CO_2\%)_t]}$ [from flue gas analysis]
Where, $(CO_2\%)_t$	=	Theoretical CO_2
$(CO_2\%)_a$	=	Actual CO_2% measured in flue gas
$(CO_2)_t$	=	$\frac{Moles\ of\ C}{Moles\ of\ N_2\ +\ Moles\ of\ C + Moles\ of\ S}$
Moles of N_2	=	$\frac{Wt\ of\ N_2\ in\ theoritical\ air}{Mol.Wt\ of\ N_2} + \frac{Wt\ of\ N_2\ in\ fuel}{Mol.Wt\ of\ N_2}$
Moles of C	=	$\frac{Wt\ of\ C\ in\ fuel}{Molecular\ Wt\ of\ C}$
Moles of S	=	$\frac{Wt\ of\ S\ in\ fuel}{Molecular\ Wt\ of\ S}$
c) Actual mass of air supplied/ kg of fuel (AAS)	=	{1 + EA/100} xTheoretical air

The various losses associated with the operation of a boiler are discussed below with required formula.

1. Heat loss due to dry flue gas

This is the greatest boiler loss and can be calculated with the following formula:

$$L_1 = \frac{m\,x\,C_p\,x\,(T_f - T_a)}{GCV\ of\ fuel}\ x\ 100$$

Where,

L_1 = % Heat loss due to dry flue gas

m = Mass of dry flue gas in kg/kg of fuel

= Combustion products from fuel: CO_2 + SO_2 + Nitrogen in fuel + Nitrogen in the actual mass of air supplied + O_2 in flue gas.
(H_2O/Water vapour in the flue gas should not be considered)

C_p = Specific heat of flue gas in kcal/kg

T_f = Flue gas temperature in ^oC

T_a = Ambient temperature in ^oC

Note-1:

For Quick and simple calculation of boiler efficiency use the following.

A: Simple method can be used for determining the dry flue gas loss as given below.

a) Percentage heat loss due to dry flue gas = $\frac{m\,x\,C_p\,x\,(T_f - T_a)}{GCV\ of\ fuel}\ x\ 100$

Total mass of dry flue gas (m)/kg of fuel = Mass of actual air supplied (ASS) + Mass of fuel supplied – (M + 9 H_2)

Note-2: Water vapour is produced from hydrogen in fuel, moisture present in fuel and air during the combustion. The losses due to these components have not been included in the dry flue gas loss since they are separately calculated as a wet flue gas loss.

2. Heat loss due to evaporation of water formed due to H_2 in fuel (%)

The combustion of hydrogen causes a heat loss because the product of combustion is water. This water is converted to steam and this carries away heat in the form of its latent heat.

$$L_2 = \frac{9 \times H_2 \times \{584 + C_p (T_f - T_a)\}}{\text{GCV of fuel}} \times 100$$

Where

H_2	=	kg of hydrogen present in fuel on 1 kg basis
C_p	=	Specific heat of superheated steam in kcal/kg ^oC
T_f	=	Flue gas temperature in ^oC
T_a	=	Ambient temperature in ^oC
584	=	Latent heat corresponding to partial pressure of water vapour in kcal/kg

3. Heat loss due to moisture present in fuel

Moisture entering the boiler with the fuel leaves as a superheated vapour. This moisture loss is made up of the sensible heat to bring the moisture to boiling point, the latent heat of evaporation of the moisture, and the superheat required to bring this steam to the temperature of the exhaust gas. This loss can be calculated with the following formula

$$L_3 = \frac{M \times \{584 + C_p (T_f - T_a)\}}{\text{GCV of fuel}} \times 100$$

where

M	=	kg of moisture in fuel in 1 kg basis
C_p	=	Specific heat of superheated steam in kcal/kg ^oC
T_f	=	Flue gas temperature in ^oC
T_a	=	Ambient temperature in ^oC
584	=	Latent heat corresponding to partial pressure of water vapour in kcal/kg

4. Heat loss due to moisture present in air

Vapour in the form of humidity in the incoming air, is superheated as it passes through the boiler. Since this heat passes up the stack, it must be included as a boiler loss.

To relate this loss to the mass of coal burnt, the moisture content of the combustion air and the amount of air supplied per unit mass of coal burnt must be known.

The mass of vapour that air contains can be obtained from psychrometric charts and typical values are included as follows:

Dry-Bulb Temp (°C)	Wet Bulb Temp (°C)	Relative Humidity (%)	Kilogram water per Kilogram dry air (Humidity Factor)
20	20	100	0.016
20	14	50	0.008
30	22	50	0.014
40	30	50	0.024

$$L_4 = \frac{AAS \times humidity\ factor \times C_p \times (T_f - T_a)}{\text{GCV of fuel}} \times 100$$

where

AAS	=	Actual mass of air supplied per kg of fuel
Humidity factor	=	kg of water/kg of dry air
C_p	=	Specific heat of superheated steam in kcal/kg°C
T_f	=	Flue gas temperature in °C
T_a	=	Ambient temperature in °C (dry bulb)

5. Heat loss due to incomplete combustion

Products formed by incomplete combustion could be mixed with oxygen and burned again with a further release of energy. Such products include CO, H_2, and various hydrocarbons and are generally found in the flue gas of the boilers. Carbon monoxide is the only gas whose concentration can be determined conveniently in a boiler plant test.

$$L_5 = \frac{\%CO \times C}{\%CO + \%CO_2} \times \frac{5654^*}{GCV\ of\ fuel} \times 100$$

L_5	=	% Heat loss due to partial conversion of C to CO
CO	=	Volume of CO in flue gas (%) (1% = 10000 ppm)
CO_2	=	Actual Volume of CO_2 in flue gas (%)
C	=	Carbon content kg / kg of fuel

* Heat loss due to partial combustion of carbon, kcal/kg of Carbon.

6. Heat loss due to radiation and convection:

The other heat losses from a boiler consist of the loss of heat by radiation and convection from the boiler casting into the surrounding boiler house.

Normally surface loss and other unaccounted losses is assumed based on the type and size of the boiler as given below

For industrial fire tube / packaged boiler = 1.5 to 2.5%

For industrial water tube boiler = 2 to 3%

For power station boiler = 0.4 to 1%

However it can be calculated if the surface area of boiler and its surface temperature are known as given below :

$$L_6 = 0.548 \times [(T_s / 55.55)^4 - (T_a / 55.55)^4] + 1.957 \times (T_s - T_a)^{1.25} \times \text{sq.rt of } [(196.85 V_m + 68.9) / 68.9]$$

where

L_6 = Radiation loss in W/m^2

V_m = Wind velocity in m/s

T_s = Surface temperature (K)

T_a = Ambient temperature (K)

Heat loss due to unburnt carbon in fly ash and bottom ash:

Small amounts of carbon will be left in the ash and this constitutes a loss of potential heat in the fuel. To assess these heat losses, samples of ash must be analyzed for carbon content. The quantity of ash produced per unit of fuel must also be known.

7. Heat loss due to unburnt in fly ash (%):

$$L_7 = \frac{\text{Total ash collected / kg of fuel burnt x G.C.V of fly ash}}{\text{GCV of fuel}} \; x \; 100$$

8. Heat loss due to unburnt in bottom ash (%)

$$L_8 = \frac{\text{Total ash collected / kg of fuel burnt x G.C.V of bottom ash}}{\text{GCV of fuel}} \; x \; 100$$

Heat Balance:

Having established the magnitude of all the losses mentioned above, a simple heat balance would give the efficiency of the boiler. The efficiency is the difference between the energy input to the boiler and the heat losses calculated.

Boiler Heat Balance:

Input/Output Parameter		kcal / kg of fuel	%
Heat Input in fuel			100
Various Heat losses in boiler			
1. Dry flue gas loss	L_1		
2. Loss due to hydrogen in fuel	L_2		
3. Loss due to moisture in fuel	L_3		
4. Loss due to moisture in air	L_4		
5. Partial combustion of C to CO	L_5		
6. Surface heat losses	L_6		
7. Loss due to Unburnt in fly ash	L_7		
8. Loss due to Unburnt in bottom ash	L_8		
Total Losses			
Boiler efficiency = $100 - (L_1+L_2+L_3+L_4+L_5+L_6+L_7+L_8)$			

1.8 Example: Boiler Efficiency Calculation

1.8.1 For Coal fired Boiler

The following are the data collected for a boiler using coal as the fuel. Find out the boiler efficiency by indirect method.

Fuel firing rate	=	5600 kg/hr
Steam generation rate	=	21940 kg/hr
Steam pressure	=	43 kg/cm^2 (g)
Steam temperature	=	377 ^{o}C
Feed water temperature	=	96 ^{o}C
%CO_2 in Flue gas	=	14
%CO in flue gas	=	0.55
Average flue gas temperature	=	190 ^{o}C
Ambient temperature	=	31 ^{o}C
Humidity in ambient air	=	0.0204 kg / kg dry air
Surface temperature of boiler	=	70 ^{o}C
Wind velocity around the boiler	=	3.5 m/s
Total surface area of boiler	=	90 m^2
GCV of Bottom ash	=	800 kcal/kg
GCV of fly ash	=	450 kcal/kg
Ratio of bottom ash to fly ash	=	90:10

Fuel Analysis (in %)		
Ash	=	48
Moisture	=	4.4
Carbon	=	36
Hydrogen	=	2.6
Nitrogen	=	1.1
Oxygen	=	7.3
Sulphur	=	0.6
GCV	=	3501 kcal/kg

Boiler efficiency by indirect method

Step – 1 Find theoretical air requirement

Theoretical air required for complete combustion = $[(11.6\,x\,C)+\{34.8\,x\,(H_2-O_2/8)\}+(4.35\,x\,S)]/100$ kg/kg of coal

= [(11.6 x36) + {34.8 x (2.6 – 7.3/8)} + (4.35 x 0.6)] / 100

= **4.79 kg / kg of coal**

Step – 2 Find theoretical CO_2 %

% CO_2 at theoretical condition $(CO_2)_t$ = $\dfrac{Moles\ of\ C}{Moles\ of\ N_2 + Moles\ of\ C + Moles\ of\ S}$

Where,

Moles of N_2 = $\dfrac{Wt\ of\ N_2\ in\ theoritical\ air}{Mol.wt\ of\ N_2} + \dfrac{Wt\ of\ N_2\ in\ fuel}{Mol.Wt\ of\ N_2}$

Moles of N_2 = $\dfrac{4.79\,x\,77/100}{28} + \dfrac{0.011}{28} = 0.1321$

Moles of C = 0.36/12 = 0.03

Moles of S = 0.006/32 = 0.0001875

$(CO_2)_t$ = $\dfrac{0.03}{0.03 + 0.1321 + 0.0001875}$

$(CO_2)_t$ = 18.48 %

Step – 3 To find Excess air supplied

Actual CO_2 measured in flue gas = 14.0%

% Excess air supplied (EA) = $\frac{7900\,x[(CO_2\%)_t - (CO_2\%)_a]}{(CO_2)_a\%\ x\ [100-(CO_2\%)_t]}$

= $\frac{7900\,x[18.48-14]}{14\ x\ [100-18.48]}$

= **31 %**

Step – 4 To find actual mass of air supplied

Actual mass of air supplied = {1 + EA/100} x theoretical air

= {1 + 31/100} x 4.79

= **6.27 kg/kg of coal**

Step –5 To find actual mass of dry flue gas

Mass of dry flue gas = Mass of CO_2 in flue gas +Mass of N_2 content in the fuel+ Mass of N_2 in the combustion air supplied + Mass of oxygen in flue gas + Mass of SO_2 in flue gas

Mass of dry flue gas = $\frac{0.36\,x\,44}{12} + 0.011 + \frac{6.27\,x\,77}{100} + \frac{(6.27-4.79)\,x\,23}{100} + \frac{0.006x\,64}{32}$

= **6.51 kg / kg of coal**

Step – 6 To find all losses

1. % Heat loss in dry flue gas (L_1) = $\frac{m\,x\,C_p\,x\,(T_f - T_a)}{GCV\ of\ fuel}\ x\ 100$

= $\frac{6.51\,x\,0.24\,x\,(190-31)}{3501}\ x\ 100$

L_1 = **7.1 %**

2. % Heat loss due to formation of water from H_2 in fuel (L_2) = $\frac{9\ x\ H_2\ x\ \{584\ +\ C_p\ (T_f\text{-}T_a\ )\}}{GCV\ of\ fuel}\ x\ 100$

= $\frac{9\ x\ H_2\ x\ \{584\ +\ C_p\ (T_f\text{-}T_a\ )\}}{GCV\ of\ fuel}\ x\ 100$

L_2 = **4.36 %**

3. % Heat loss due to moisture in fuel (L_3) = $\frac{M\ x\ \{584\ +\ C_p\ (T_f\text{-}T_a\ )\}}{GCV\ of\ fuel}\ x\ 100$

= $\frac{0.044\ x\ \{584\ +\ 0.43\ (190\text{-}31\ )\}}{3501}\ x\ 100$

L_3 = **0.82 %**

4. % Heat loss due to moisture in air (L_4)

$$= \frac{AAS \; x \; humidity \; x \; C_p \; x \; (T_f - T_a)}{GCV \; of \; fuel} \; x \; 100$$

$$= \frac{6.27 \; x \; 0.0204 \; x \; 0.43 \; x \; (190 - 31)}{3501} \; x \; 100$$

L_4 = **0.25 %**

5. % Heat loss due to partial conversion of C to CO (L_5)

$$= \frac{\%CO \; x \; C}{\%CO + \%CO_2} \; x \; \frac{5654}{GCV \; of \; fuel} \; x \; 100$$

$$= \frac{0.55 \; x \; 0.36}{0.55 + 14} \; x \; \frac{5654}{3501} \; x \; 100$$

L_5 = **2.2 %**

6. Heat loss due to radiation and convection (L_6)

= 0.548 x [$(343/55.55)^4$ – $(304/55.55)^4$] + 1.957 x $(343 - 304)^{1.25}$ x sq.rt of [(196.85 x 3.5 + 68.9) / 68.9]

= 937.62 w/m^2

= 937.62 x 0.86

= 806.35 kcal / m^2

Total radiation and convection loss per hour = 806.35 x 90

= 72571.6 kcal/hr

% radiation and convection loss = $\frac{72571.6 \; x \; 100}{3501 \; x \; 5600}$

L_6 = **0.37 %**

7. % Heat loss due to unburnt in fly ash

% Ash in coal	=	48
Ratio of bottom ash to fly ash	=	90:10
GCV of fly ash	=	450 kcal/kg
Amount of fly ash in 1 kg of coal	=	0.1 x 0.48
	=	0.048 kg
Heat loss in fly ash	=	0.048 x 450
	=	21.6 kcal / kg of coal
% heat loss in fly ash	=	21.6 x 100 / 3501
L_7	=	**0.62 %**

8. % Heat loss due to unburnt in bottom ash

GCV of bottom ash	=	800 kcal/kg
Amount of bottom ash in 1 kg of coal	=	0.9 x 0.48
	=	0.432 kg
Heat loss in bottom ash	=	0.432 x 800
	=	345.6 kcal/kg of coal
% Heat loss in bottom ash	=	345.6 x 100 / 3501
L_8	=	**9.87 %**

Boiler efficiency by indirect method	=	$100 – (L_1+ L_2+ L_3+ L_4+ L_5+ L_6+ L_7+ L_8)$
	=	100-(7.1 + 4.36 + 0.82 + 0.25 + 2.2 + 0.37 + 0.62 + 9.87)
	=	100-25.59
	=	**74.41 %**

Summary of Heat Balance for Coal Fired Boiler

Input/Output Parameter		kcal / kg of coal	% loss
Heat Input		3501	100
Losses in boiler			
1. Dry flue gas	L_1	248.6	7.1
2. Loss due to hydrogen in fuel	L_2	152.6	4.36
3. Loss due to moisture in fuel	L_3	28.7	0.82
4. Loss due to moisture in air	L_4	8.7	0.25
5. Partial combustion of C to CO	L_5	77.0	2.2
6. Surface heat losses	L_6	13.0	0.37
7. Loss due to Unburnt in fly ash	L_7	21.7	0.62
8. Loss due to Unburnt in bottom ash	L_8	345.5	9.87
Boiler Efficiency = $100 – (L_1 + L_2+ L_3+ L_4+ L_5+ L_6+ L_7+ L_8)$ = 74.41 %			

1.8.2 Efficiency for an oil fired boiler

The following are the data collected for a boiler using furnace oil as the fuel. Find out the boiler efficiency by indirect method.

Ultimate analysis (%)

Carbon	=	84
Hydrogen	=	12
Nitrogen	=	0.5
Oxygen	=	1.5
Sulphur	=	1.5

Moisture	=	0.5
GCV of fuel	=	10000 kcal/kg
Fuel firing rate	=	2650 kg/hr
Surface Temperature of boiler	=	80 ^{o}C
Surface area of boiler	=	90 m^2
Humidity	=	0.025 kg/kg of dry air
Wind speed	=	3.8 m/s

Flue gas analysis (%)		
Flue gas temperature	=	190 ^{o}C
Ambient temperature	=	30 ^{o}C
CO_2% in flue gas by volume	=	10.8
O_2% in flue gas by volume	=	7.4

a) Theoretical air required	=	$[(11.6\,x\,C)+\{34.8\,x\,(H_2-O_2/8)\}+(4.35\,x\,S)]/100$ kg/kg of fuel. [from fuel analysis]
	=	[(11.6 x 84) + [{34.8 x (12 – 1.5/8)} + (4.35 x 1.5)] / 100
	=	**13.92 kg/kg of oil**
b) Excess Air supplied (EA)	=	$\frac{O_2\%}{21-O_2\%} x100$ [from flue gas analysis]
	=	$\frac{7.4}{21-7.4} x100$
	=	**54.4 %**
c) Actual mass of air supplied/ kg of fuel (AAS)	=	{1 + EA/100} x theoretical air
	=	{1 + 54.4/100} x 13.92
	=	21.49 kg / kg of fuel
Mass of dry flue gas	=	Mass of (CO_2 + SO_2 + N_2 + O_2) in flue gas + N_2 in air we are supplying.
Mass of dry flue gas	=	$\frac{0.84\,x\,44}{12}+\frac{0.015\,x\,64}{32}+0.005+\frac{(21.49-13.92)\,x\,23}{100}+\frac{21.49\,x\,77}{100}$
	=	**21.4 kg / kg of oil**

% Heat loss in dry flue gas	=	$\frac{m \, x \, C_p \, x \, (T_f - T_a)}{GCV \; of \; fuel} \; x \; 100$
	=	$\frac{21.4 \, x \, 0.24 \, x \, (190 - 30)}{10000} \; x \; 100$
L_1	=	**8.22 %**

Heat loss due to evaporation of water due to H_2 in fuel (%)	=	$\frac{9 \text{ x } H_2 \text{ x } \{584 + C_p \, (T_f - T_a)\}}{\text{GCV of fuel}} \; x \; 100$
	=	$\frac{9 \text{ x } 0.12 \text{ x } \{584 + 0.43 \, (190\text{-}30)\}}{10000} \; x \; 100$
L_2	=	**7.05 %**
% Heat loss due to moisture in fuel	=	$\frac{M \text{ x } \{584 + C_p \, (T_f - T_a)\}}{\text{GCV of fuel}} \; x \; 100$
	=	$\frac{0.005 \text{ x } \{584 + 0.43 \, (190\text{-}30)\}}{10000} \; x \; 100$
L_3	=	**0.033%**

% Heat loss due to moisture in air	=	$\frac{AAS \, x \, humidity \; factor \, x \; C_p \text{ x } (T_f - T_a)}{\text{GCV of fuel}} \; x \; 100$
	=	$\frac{21.4 \, x \, 0.025 \, x \; 0.43 \text{ x } (190\text{-}30)}{10000} \; x \; 100$
L_4	=	**0.37 %**
Radiation and convection loss (L_5)	=	$0.548 \text{ x } [(T_s / 55.55)^4 - (T_a / 55.55)^4] + 1.957 \text{ x } (T_s - T_a)^{1.25}$ x sq.rt of $[(196.85 \, V_m + 68.9) / 68.9]$
	=	$0.548 \text{ x } [(353 / 55.55)^4 - (303 / 55.55)^4] + 1.957 \text{ x } (353 - 303)^{1.25}$ x sq.rt of $[(196.85 \text{ x } 3.8 + 68.9) / 68.9]$
	=	1303 W/m^2
	=	1303 x 0.86
	=	1120.58 kcal / m^2
Total radiation and convection loss per hour	=	1120 .58 x 90 m^2
	=	100852.2 kcal

% Radiation and convection loss	=	$\frac{100852.2}{10000 \, x \, 2650} \, x \, 100$
L_5	=	**0.38 %** Normally it is assumed as 0.5 to 1 % for simplicity

Boiler efficiency by indirect method	=	$100 - (L_1 + L_2 + L_3 + L_4 + L_5)$
	=	100-(8.22 + 7.05 + 0.033 + 0.37 + 0.38)
	=	100 – 16.05
	=	**83.95 %**

Summary of Heat Balance for the Boiler Using Furnace Oil

Input/Output Parameter		**kcal / kg of furnace oil**	**%Loss**
Heat Input		10000	100
Losses in boiler :			
1. Dry flue gas, L_1	L_1	822	8.22
2. Loss due to hydrogen in fuel, L_2	L_2	705	7.05
3. Loss due to Moisture in fuel, L_3	L_3	3.3	0.033
4. Loss due to Moisture in air, L_4	L_4	37	0.37
5. Surface heat losses, L_5	L_5	38	0.38
Boiler Efficiency = 100 – (L_1 + L_2+ L_3+ L_4+ L_5) = 83.95 %			

Note: Simple method can be used for determining the dry flue gas loss as given below.

Percentage heat loss due to dry flue gas = $\frac{m \, x \, C_p \, x \, (T_f - T_a)}{GCV \; of \; fuel} \; x \; 100$

Total mass of dry flue gas (m)

= Total mass of wet flue gas – Mass of water vapour in flue gas

= Mass of actual air supplied (AAS) + Mass of fuel supplied – (M + 9 H_2)

= 21.49 + 1 – {0.005 + (9 x 0.12)} = 21.4

% Dry flue gas loss = $\frac{21.4 \, x \, 0.24 \, x \, (190 - 30)}{10000} \; x \; 100 \;$ = 8.22%

1.9 Factors Affecting Boiler Performance

The various factors affecting the boiler performance are listed below:

- Periodical cleaning of boilers
- Periodical soot blowing

- Proper water treatment programme and blow down control
- Draft control
- Excess air control
- Percentage loading of boiler
- Steam generation pressure and temperature
- Boiler insulation
- Quality of fuel

All these factors individually/combined, contribute to the performance of the boiler and reflected either in boiler efficiency or evaporation ratio. Based on the results obtained from the testing further improvements have to be carried out for maximizing the performance. The test can be repeated after modification or rectification of the problems and compared with standard norms. Energy auditor should carry out this test as a routine manner once in six months and report to the management for necessary action.

1.10 Boiler Terminology

MCR: Steam boilers rated output is also usually defined as MCR (Maximum Continuous Rating). This is the maximum evaporation rate that can be sustained for 24 hours and may be less than a shorter duration maximum rating

Boiler Rating

Conventionally, boilers are specified by their capacity to hold water and the steam generation rate. Often, the capacity to generate steam is specified in terms of equivalent evaporation (kg of steam / hour at 100^{o}C). Equivalent evaporation - "from and at" 100^{o}C. The equivalent of the evaporation of 1 kg of water at 100^{o}C to steam at 100^{o}C.

Primary air: That part of the air supply to a combustion system which the fuel first encounters.

Secondary air: The second stage of admission of air to a combustion system, generally to complete combustion initiated by the primary air. It can be injected into the furnace of a boiler under relatively high pressure when firing solid fuels in order to create turbulence above the burning fuel to ensure good mixing with the gases produced in the combustion process and thereby complete combustion

Absolute pressure: The sum of the gauge and the atmospheric pressure. For instance, if the steam gauge on the boiler shows 9 kg/cm^{2}g the absolute pressure of the steam is 10 kg/cm^2(a).

Solved Example:

The steam requirement of an export oriented unit (EOU) is met by a 6 TPH oil fired package boiler generating steam at 10 kg/cm^2. The monthly steam consumption of the unit is 3000 tonnes.

Other data are given below:

Fuel oil composition: Carbon = 86%; Hydrogen = 12%; Oxygen= 0.5%; Sulphur =1.5%

Specific heat of flue gases, Cp = 0.27 kcal/kg^{o}C

G.C.V. of fuel oil	= 10,000 kcal/kg
Sp. heat of super-heated water vapour	= 0.45 kcal/kg^{o}C
Enthalpy of steam at 10 kg/cm^2	= 665kcal/kg
Feed water temperature	= 85 ^{o}C
% O_2 in dry flue gas	= 6%
Flue gas temperature at boiler outlet	= 240 ^{o}C
Ambient temperature	= 30^{o}C
Cost of fuel oil	= Rs.43 per kg
Radiation and other unaccounted losses	= 2.45%

The export oriented unit is costing its steam cost based on the fuel consumption cost with additional 15% to account for the auxiliary and consumables.

A neighbouring continuous process plant now offers to supply the required steam at 10 kg/cm^2 to the export oriented unit at a cost of Rs 3300 per tonne with a condition that all the condensate will be returned back.

Calculate the following:

a) Boiler efficiency

b) Cost advantage per tonne of availing steam from neighbouring plant in place of in-house generation and also monthly monetary saving.

Solution:

First calculate the efficiency of Boiler (in EOU)

Theoretical air required = 11.6 C + 34.8 (H_2 – O_2/8) + 4.35 S
= [11.6 X 86 + 34.8 (12 – 0.5/8) + 4.35 X 1.5] x 1/100
= 14.195 = Say 14.2

% Excess Air = [% O_2 / (21 - % O_2)] X 100
= [6 / (21 – 6)] X 100 = 40%

AAS = Actual amount of air supplied = 14.2 X 1.4
= **19.88 kg per kg of fuel oil**

Mass of dry flue gas m_{dfg} = Mass of combustion gases due to presence of C,H,S +Mass of N_2 supplied
= (0.86 X 44/12) + (0.015 X 64 / 32) + [(19.88 – 14.2) X 23 / 100] + (19.88 X 77/100)
= 19.797

Mass dry flue gas, say = 19.8 kg / kg fuel

or

Alternatively mass of dry flue gas = (AAS + 1) – 9 H_2
= (19.88 + 1) – 9 X 0.12 = 19.8 kg/kg fuel

L_1 = % heat loss in dry flue gas = $[m_{dfg} \times Cp \times (T_q - T_a) / GCV] \times 100$

$$= \frac{19.8 \times 0.27 \times (240 - 30)}{10{,}00} \times 100$$

= **11.23%**

L_2 = Loss due to presence of hydrogen forming water vapour

$$= \frac{9 \times H \ [584 + Cps\ (T_g - T_a)]}{GCV} \times 100$$

$$= \frac{9 \times 0.12\ [584 + 0.45\ (240 - 30)]}{10000} \times 100$$

= 7.33%

L_3 = Radiation and other unaccounted losses **= 2.45%**

Total losses = $L_1 + L_2 + L3$

= 11.23 + 7.33 + 2.45 = **21.05 %**

Efficiency of the EOU boiler by indirect method

= 100 – 21.05 =78.99 %

= **Say 79 %**

<u>Secondly calculate the cost of steam in the EOU plant</u>

Evaporation Ratio = $[(\eta X\ GCV) / (h_g - h_f)] \times 100$

= $[(0.79 \times 10000) / (665 - 85)] \times 100$

= 13.62 kg Steam / kg Fuel

Fuel oil consumption = 1000 / 13.62 kg per tonne of steam

Fuel oil consumption = 73.42 kg/tonne of steam gen

Cost of fuel oil = Rs. 43 per kg

Cost of steam in EOU = Fuel cost + 15% fuel cost

= 73.42 x 1.15 x 43

= Rs.3,599 per tonne

Say = Rs 3600 per tonne

Selling cost of steam from neighbouring plant = Rs 3300 per tonne

Cost advantage = 3600 – 3300

= Rs.300 per tonne

Annual Savings = Rs.300 per tonne x 3000 tonne/month x 12 month

= Rs. 108 Lakhs

QUESTIONS	
S-1	Define boiler efficiency.
S-2	Why is boiler efficiency by indirect method more useful than direct method?
S-3	What instruments are required for indirect efficiency testing?
S-4	What is the difference between dry flue gas loss and wet flue gas loss?
S-5	Which is the best location for sampling flue gas analysis?
S-6	What is excess air? How to determine excess air if oxygen / carbon dioxide percentage is measured in the flue gas?
S-7	As a means of performance evaluation, explain the difference between efficiency and evaporation ratio.
S-8	Testing coal-fired boiler is more difficult than oil-fired boiler. Give reasons.
S-9	What are controllable and uncontrollable losses in a boiler?
S-10	State two causes for rise in exit flue gas temperature in a boiler.
L-1	From the fuel analysis given in the problem 1.8.1. calculate the LCV of the fuel
L-2	List out different temperatures to be measured during a boiler audit?
N-1	The following are the data collected for a boiler using furnace oil as the fuel. Determine the boiler efficiency based on GCV by indirect method ignoring radiation and convection losses. Ultimate chemical analysis (% weight): Carbon : 84, Hydrogen : 12, Nitrogen : 0.5, Oxygen : 1.5, Sulphur: 1.5, Moisture : 0.5, NCV of fuel 9,763 kcal/kg and humidity 0.025 kg moisture /kg of dry air. Flue gas analysis: CO_2 : 9.8% volume, flue gas exit temperature : 190^{o}C and ambient temperature : 30^{o}C.

REFERENCE

1. Energy audit Reports of National Productivity Council
2. Energy Hand book, Second edition, Von Nostrand Reinhold Company - Robert L.Loftness
3. Industrial boilers, Longman Scientific Technical 1999

www.boiler.com
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Chapter 2

Energy Performance Assessment of Furnaces

Learning Objectives

In this chapter you will learn about

- Furnace performance: terms and definitions
- Furnace heat balance method: efficiency evaluation
- Factors affecting furnace performance
- Mean specific heat calculation

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Conduct furnace efficiency test**
- ✓ **Perform heat balance calculation**

2. ENERGY PERFORMANCE ASSESSMENT OF FURNACES

2.1 Industrial Heating Furnaces

A heating furnace is by definition a device for heating materials and therefore a user of energy. Heating furnaces can be divided into batch-type (Job at stationary position) and continuous type (large volume of work output at regular intervals). The types of batch furnace include box, bogie, cover, etc. For mass production, continuous furnaces are used in general. The types of continuous furnaces include pusher-type furnace (Figure 2.1), walking hearth-type furnace, rotary hearth and walking beam-type furnace. (Figure 2.2)

The primary energy required for reheating / heat treatment (say annealing) furnaces are in the form of Furnace oil, LSHS, LDO or electricity

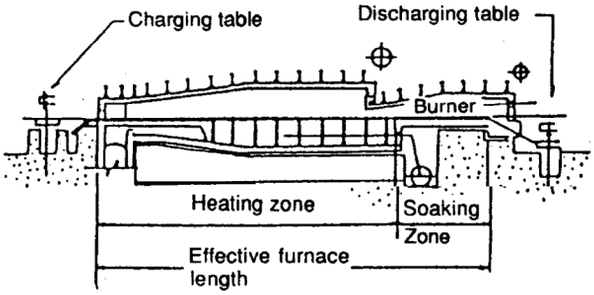

Figure 2.1: Pusher-Type 3-Zone Reheating Furnace

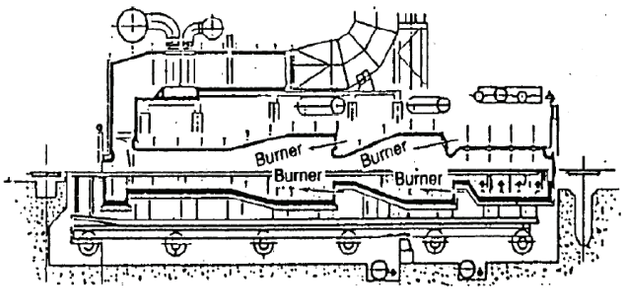

Figure 2.2: Walking Beam Reheating Furnace

2.2 Purpose of the Performance Test

- To make a heat balance of the furnace
- To find out the efficiency of the furnace
- To find out the specific energy consumption

The purpose of the performance test is to determine various losses, efficiency of the furnace and specific energy consumption for comparing with design values or best practice norms. There are many factors affecting furnace performance such as capacity utilization of furnaces, excess air ratio, final heating temperature etc. Performance test is the key for assessing current level of performances and finding the scope for improvements and productivity.

2.3 Reference Standards

In addition to conventional methods, Japanese Industrial Standard (JIS) GO702 “Method of heat balance for continuous furnaces for steel” is used for the purpose of establishing the heat losses and efficiency of reheating furnaces. The standard has been simplified to suit practical calculation in the industry.

2.4 Performance Terms and Definitions

1. Furnace Efficiency, η $= \dfrac{Heat\ output}{Heat\ input} x100$

$= \dfrac{Heat\ in\ stock\ (material), kcal}{Heat\ in\ fuel\ /\ electricity, kcal} x100$

2. Specific Energy Consumption $= \dfrac{Amount\ of\ energy\ consumed, kcal\ /\ hr}{Quantity\ of\ material\ processed, T\ /\ hr}$

2.5 Furnace Heat Balance Method

Heat balance helps us to numerically understand the present heat loss and efficiency and improve the furnace operation using these data. Thus, preparation of heat balance is a pre-requirement for assessing energy conservation potential. The methodology of a typical furnace heat balance is given simultaneously along with an example 2.6.

The total heat input is provided in the form of fuel or power. The desired output is the heat supplied for heating the material or process. Other heat outputs in the furnaces are undesirable heat losses. The inputs and outputs are calculated on the basis of per tonne of stock/charge.

The major losses that occur in the fuel fired furnace (Figure 2.3) are listed below.

1. Heat lost through exhaust gases either as sensible heat or as incomplete combustion
2. Heat loss through furnace walls and hearth
3. Heat loss to the surroundings by radiation and convection from the outer surface of the walls
4. Heat loss through gases leaking through cracks, openings and doors.

Similar to the method of evaluating boiler efficiency by indirect method, furnace efficiency can also be calculated from heat balance. Furnace efficiency is calculated after subtracting sensible heat loss in flue gas, loss due to moisture in flue gas, heat loss due to openings in furnace, heat loss through furnace skin and other unaccounted losses from the heat input to the furnace.

In order to carry out a heat balance, various parameters that are required are hourly oil consumption, material output, excess air quantity, temperature of flue gas, temperature of furnace at various zones,

skin temperature and hot combustion air temperature.

The energy absorbed by the material requires the use of specific heat which can be obtained from reference manual/data book.

If the process requires a change in state, from solid to liquid, or liquid to gas, then in addition to sensible heat, an additional quantity of energy is required called the latent heat of fusion or latent heat of evaporation and this quantity of energy needs to be added to the total energy requirement.

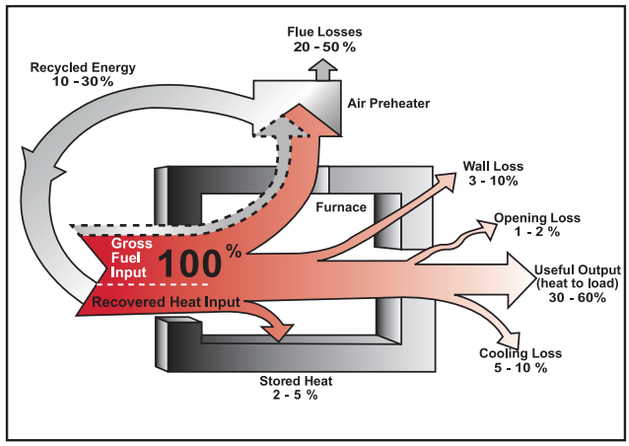

Figure 2.3 Fuel Fired Furnace

Measurement Parameters

The following are some of the key measurements to be made for working out the energy balance in oil fired reheating furnaces.

i) Weight of stock / Number of billets heated
ii) Temperature of the stocks/billets
iii) Temperature of furnace walls, roof etc
iv) Area of furnace walls, roof etc
v) Flue gas temperature
vi) Flue gas analysis
vii) Fuel oil consumption

Instruments like infrared thermometer, fuel consumption monitor, surface thermocouple and other measuring devices are required to measure the above parameters. Reference manual should be referred for data like specific heat, humidity etc.

Furnace Efficiency

The efficiency of a furnace is the ratio of useful heat output to heat input. The direct determination of furnace efficiency is carried out as follows.

$$Thermal\ efficiency\ of\ the\ furnace = \frac{Heat\ in\ the\ stock}{Heat\ in\ the\ fuel\ consumed}$$

The quantity of heat to be imparted (Q) to the stock can be found from the formula

$$Q = m\,x\,C_p\,x\,(t_2 - t_1)$$

Where,

Q	=	Quantity of heat in kcal
m	=	Weight of the material in kg
C_p	=	Mean specific heat, kcal/kg ^oC
t_2	=	Final temperature desired, ^oC
t_1	=	Initial temperature of the charge before it enters the furnace, ^oC

2.6 Example: Heat Balance of Furnace

An oil-fired reheating furnace has an operating temperature of around 1340^{o}C. Average fuel consumption is 400 litres/hour. The flue gas exit temperature after air preheater is 655 ^{o}C. Air is preheated from ambient temperature of 40 ^{o}C to 190 ^{o}C through an air pre-heater. The furnace has 460 mm thick wall (x) on the billet extraction outlet side, which is 1 m high (D) and 1 m wide. Draw a heat balance to identify heat losses, efficiency and specific fuel consumption. The other data are as follows.

Flue gas temperature after air preheater	=	655 ^{o}C
Ambient temperature	=	40 ^{o}C
Abs. humidity	=	0.03437 kg/kg dry air
Preheated air temperature	=	190 ^{o}C
Specific gravity of oil	=	0.92
Average fuel oil consumption	=	400 Litres / hr
	=	400 x 0.92 =368 kg/hr
O_2 in flue gas	=	12%
CO_2 in flue gas	=	6.5%
CO in flue gas	=	50 ppm
Weight of stock	=	6000 kg/hr
Specific heat of Billet	=	0.12 kcal/kg ^{o}C
Surface temperature of ceiling	=	85 ^{o}C
Surface temperature of side walls	=	100 ^{o}C
Surface temperature of flue duct	=	64 ^{o}C
Area of ceiling	=	15 m^2
Area of side walls	=	36 m^2
Area of flue duct	=	10.3 m^2
Diameter of flue duct	=	0.4 m

Furnace oil constituents (% by weight)

Carbon - 85.9 %, Hydrogen -12%, Oxygen - 0.7%, Nitrogen - 0.5%, Sulphur - 0.5%, H_2O - 0.35%, Ash - 0.05%, GCV- 10,000 kcal/kg

Solution:

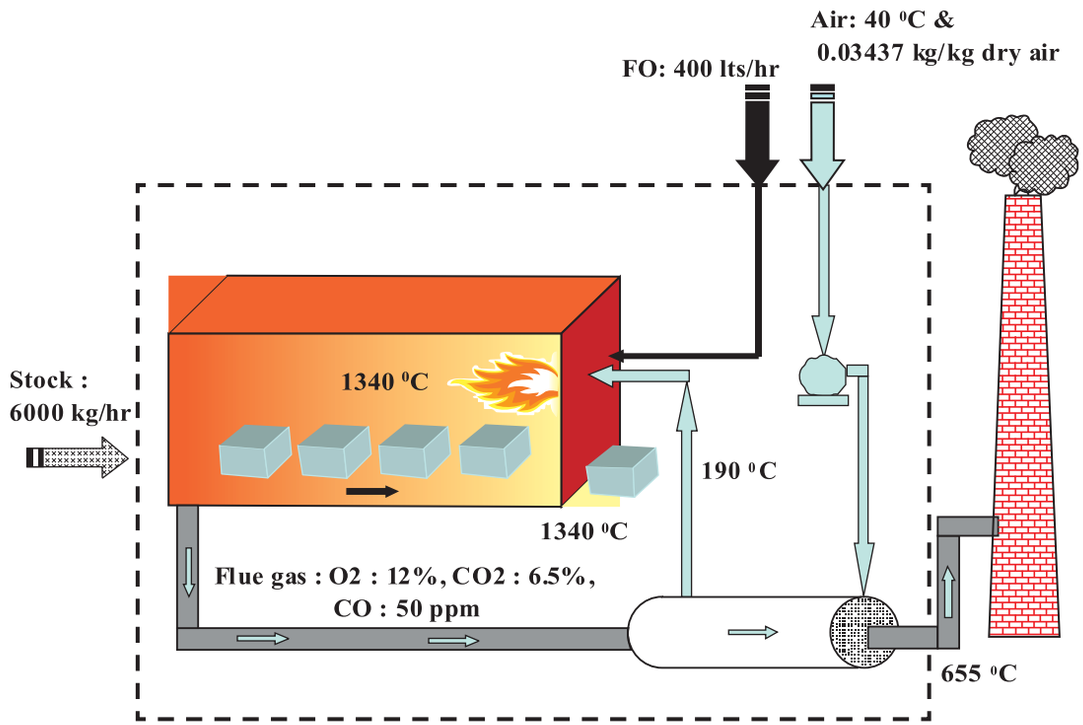

Figure 2.4: Schematic and Problem Definition

1) Calculation of air quantity and specific fuel consumption

Theoretical air required for combustion = $[(11.6\,x\,C)+\{34.8\,x\,(H_2-O_2/8)\}+(4.35\,x\,S)]/100$ kg/kg of fuel. [from fuel analysis]

= [(11.6x 85.9)+ {34.8 x (12 – 0.7/8)} + (4.35 x 0.5)]/100

= **14.12 kg of air / kg of fuel**

% Excess Air supplied (EA) = $\frac{O_2\%}{21-O_{2\%}} x100$ [from flue gas analysis]

= 12 / (21 – 12)

= **133.3 %**

Actual mass of air supplied/ kg of fuel (AAS) = {1 + EA/100} x theoretical air

= {1 + (133.3/100)} X 14.12

AAS = **32.94 kg of air / kg of fuel**

Mass of dry flue gas (m_d) = Mass of CO_2 + Mass of N_2 content in the fuel + Mass of sulphur-di-oxide + Mass of N_2 in the combustion air supplied + Mass of oxygen in flue gas

$$= \frac{0.859 x 44}{12} + 0.005 + \frac{0.005 x 64}{32} + \frac{32.94 x 77}{100} + \frac{(32.94 - 14.12) x 23}{100}$$

= **32.86 kg / kg of coal**

or

Amount of wet flue gas = AAS + 1

= **32.94 + 1 = 33.94 kg of flue gas / kg of fuel**

Amount of water vapour in flue gas (m_w) = M + 9 H_2 (M - % moisture in fuel; H_2 - % hydrogen in fuel)

= (0.35/100) + 9 x (12/100)

= **1.084 kg of H_2O/kg of fuel**

Amount of dry flue gas = Amount of wet flue gas – Amount of water vapour in flue gas

= 33.94 – 1.084

= **32.86 kg /kg of fuel**

Specific fuel consumption (F) = Amount of fuel consumed (kg/hr) / Amount of billet (t/hr)

= 368 / 6

= **61.33 kg of fuel / tonne of billet**

2) Calculation of Heat Input

Combustion heat of fuel (Q_1) = *Amount of fuel consumed per tonne of billet X GCV of fuel*

= 61.3 X 10,000

Q_1 = **6,13,300 kcal/tonne of billet**

Sensible heat of fuel (Q_2) = $F \times C_{p\ fuel} [t_f - t_a]$

Where,

Q_2 = Sensible heat of fuel in kcal/t

F = Amount of fuel consumed (kg fuel /tonne of billet)

$C_{p\ fuel}$ = Average specific heat of fuel (kcal/kg ^oC)

t_f = Fuel temperature before fed into the burner ^oC = 100 ^oC

t_a = Ambient temperature (^oC) = 40 ^oC

Q_2 = 61.33 x 0.5 x (100 – 40)

= **1840 kcal/tonne of billet**

Total heat input = $Q_1 + Q_2$
= **6,13,300 + 1840**
= **6,15,140 kcal/tonne of billet**

3) Calculation of Heat Output

Heat carried away by 1 tonne of billet (Q_3) = *1000 kg/t x C_p x ($T_0 - T_i$)*

Where,

C_p = Specific heat of billet in kcal/kg °C
T_0 = Temperature of billet at furnace exit in °C = 1340 °C
T_i = Temperature of billet at furnace entrance °C = 40 °C
= 1000 x 0.12 x (1340 – 40)
Q_3 = **1,56,000 kcal/tonne of billet**

Heat loss in dry flue gas per tonne of billet (Q_4) = *F x m x $C_{p\,fg}$ [$t_1 - t_a$]*

Where,

F = Amount of fuel consumed per ton of billet in kg fuel /tonne of billet
m = Actual mass of dry flue gas in kg/kg of fuel
$C_{p\,fg}$ = Mean specific heat of flue gas in kcal/kg °C = 0.26 kcal/kg °C (calculated in para 2.9*)*
t_1 = Temperature of flue gas after air preheater °C = 655 °C
t_a = Base temperature °C = 40 °C

Q_4 = 61.33 x 32.86 x 0.26 x [655 – 40]
= **3,22,247 kcal/tonne of billet**

Heat loss due to formation of water vapour from fuel per tonne of billet (Q_5) = *F x m_w x{584 + $C_{p\ of\ super\ heated\ vapour}$ x [$t_1 - t_a$]}*
($C_{p\ of\ super\ heated\ vapour}$ to be taken from 2.9 Table – II)

= *61.33 x 1.084 x {584 + 0.47 x (655– 40)}*
= **58,042 kcal/tonne of billet**

Heat loss due to moisture in combustion air (Q_6) = *F x AAS x Humidity of air x $C_{p\ of\ super\ heated\ vapour}$ x [$t_1 - t_a$]*

= 61.33 x 32.94 x 0.03437 x 0.47 x (655 – 40)}
= **20,070 kcal/tonne of billet**

% Heat loss due to partial conversion of C to CO (Q_7) = $F \times \frac{\%CO \times C}{\%CO + \%CO_2} \times 5654$

= $61.33 \times \frac{0.005 \times 0.859}{0.005 + 6.5} \times 5654$

= **229 kcal/tonne of billet**

Amount of heat loss from the furnace body and other sections (Q_8) = *($q_1 + q_2 + q_3 + q_4$), kcal/hr / Amount of billet (t/hr)*

Where,

q_1 = Heat loss from the furnace body ceiling surface (horizontal surface facing upward)

q_2 = Heat loss from the furnace body sidewall surfaces (vertical surfacing sideways)

q_3 = Bottom (horizontal surface facing downward)

q_4 = Heat loss from the flue gas duct between the furnace exit and the air preheater (including heat loss from the external surface of the air preheater)

q_1 = $\{h \times \Delta t^{1.25} \times A_i\} + \{4.88 \times \varepsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$

Where,

h = Natural convection heat transfer rate 2.8 (kcal/m^2 h ^{o}C)

T_w = External temperature of ceiling = 358 ^{o}K

T_a = Room temperature = 313 ^{o}K

Δt = $T_w - T_a$ = 358 – 313 = 45

A_i = Ceiling surface area = 15 m^2

ε = Emissivity of the furnace body surface (0.75)

q_1 = $\{2.8 \times 45^{1.25} \times 15\} + \{4.88 \times 0.75 \times [(358/100)^4 - (313/100)^4] \times 15\}$

= **8644 kcal/hr**

q_2 = $\{h \times \Delta t^{1.25} \times A_i\} + \{4.88 \times \varepsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$

Where,

h = Natural convection heat transfer rate 2.2 (kcal/m^2 h ^{o}C)

T_w = External temperature of side walls in ^{o}K = 373 ^{o}K

T_a = Room temperature in ^{o}K = 313 ^{o}K

Δt = $T_w - T_a$ = 373 – 313 = 60

A_i = Side wall surface area = 36 m^2

ε = Emissivity of the furnace body surface (0.75)

q_2 = $\{2.2 \times 60^{1.25} \times 36\} + \{4.88 \times 0.75 \times [(373/100)^4 - (313/100)^4] \times 36\}$

= **26084 kcal/hr**

q_3 = Bottom (horizontal surface facing downward)

As the bottom surface is not exposed to the atmosphere, q_3 is ignored in this calculation

q_4 = $\{h \times \Delta t^{1.25} / D^{0..25} \times Ai\} + \{4.88 \times \varepsilon \times [(T_w/100)^4 - (T_a/100)^4] \times A_i\}$

Where,

h = Natural convection heat transfer rate 1.1 (kcal/m^2 h ^{o}C)

T_w = External temperature of flue gas duct in ^{o}K = 337 ^{o}K

T_a = Room temperature in ^{o}K = 313 ^{o}K

Δt = $T_w - T_a$ = 337 – 313 = 24 ^{o}K

A_i = External surface area of flue gas duct = 10.3 m^2

D = Outside diameter of the flue gas duct in m = 0.4 m

ε = Emissivity of the furnace body surface (0.75)

q_4 = $[1.1 \times 24^{1.25}/0.4^{0.25} \times 10.3] + \{4.88 \times 0.75 \times [(337/100)^4 – (313/100)^4] \times 10.3\}$

= 2001 kcal/hr

$\mathbf{Q_8}$ **= ($q_1 + q_2 + q_3 + q_4$), kcal/hr / Amount of billet (t/hr)**

= (8644 + 26084 + 0 + 2001)/6

= 6,122 kcal/t

Radiation heat loss through furnace openings (Q_9) = $hr \times A \times \varphi \times 4.88 [(T_f/100)^4 – (T_0/100)^4] / t$

Where,

hr = Open time during the period of heat balancing

A = Area of an opening in m^2 = 1 m^2

φ (from table) = Co-efficient based on the profile of furnace openings

= Diameter (or) the shortest side / wall thickness = 1 / 0.46 = 2.17

= 0.70 (value corresponding to 2.17 and square shape from para 2.9 table - I)

T_f = Furnace temperature in ^{o}K = 1613 ^{o}K

T_0 = Base temperature in ^{o}K = 313 ^{o}K

t = Amount of billet in t/hr

$\mathbf{Q_9}$ = $1 \times 1 \times 0.70 \times 4.88 \times [(1613/100)^4 – (313/100)^4] / 6$

= 38,485 kcal/t

Other types of heat loss / unaccounted losses (Q_{10}) = Other types of heat loss will include the following,

- Heat carried away by cooling water in the flue damper

- Heat carried away by cooling water at the furnace access door
- Radiation from the furnace bottom
- Heat accumulated by refractory
- Instrumental error and measuring error
- Others

Q_{10} ***(unaccounted losses)*** = $(Q_1 + Q_2) - (Q_3 + Q_4 + Q_5 + Q_6 + Q_7 + Q_8 + Q_9)$
= (6,13,300 + 1,840) - (1,56,000 + 3,22,247 + 58,042+ 20,070 + 229 + 6,122 + 38,485)
= **13,945 kcal/t**

Total heat output = $Q_3 + Q_4 + Q_5 + Q_6 + Q_7 + Q_8 + Q_9 + Q_{10}$
1,56,000 + 3,22,247 + 58,042 + 20,070 + 229 + 6,122 + 38,485 + 13,945
= **6,15,140 kcal/t**

4) Heat balance table

Heat Input			**Heat output**		
Item	**kcal/t**	**%**	**Item**	**kcal/t**	**%**
Combustion heat of fuel (Q_1)	6,13,300	99.70	Heat carried away by 1 tonne of billet (Q_3)	1,56,000	25.4
Sensible heat of fuel (Q_2)	1,840	0.30	Heat loss in dry flue gas per tonne of billet (Q_4)	3,22,247	52.4
			Heat loss due to formation of water vapour from fuel per tonne of billet (Q_5)	58,042	9.4
			Heat loss due to moisture in combustion air (Q_6)	20,070	3.3
			% Heat loss due to partial conversion of C to CO (Q_7)	229	0.04
			Amount of heat loss from the furnace body and other sections (Q_8)	6,122	1.0
			Radiation heat loss through furnace openings (Q_9)	38,485	6.2
			Unaccounted losses (Q_{10})	13,945	2.3
Total	**6,15,140**	**100**		**6,15,140**	**100**

5) Efficiency of furnace

$$\eta_{furnace} = \frac{\text{Heat carried away by billet in kcal/t}}{\text{Combustion heat of fuel in kcal/t}} \times 100$$

$$= (1,56,000 / 6,13,300) \times 100$$

$$= 25.4\ \%$$

2.7 Factors Affecting Furnace Performance

The important factors, which affect the efficiency, are listed below for critical analysis.

- Under loading due to poor hearth loading and improper production scheduling
- Improper design
- Use of inefficient burner
- Insufficient draft/chimney
- Absence of waste heat recovery
- Absence of instruments/controls
- Improper operation/maintenance
- High stack loss
- Improper insulation /refractories

2.8 Useful Information

Radiation Heat Transfer

Heat transfer by radiation is proportional to the fourth power of absolute temperature. Consequently the radiation losses increase considerably as temperature increases.

°C1	°C2	K1 (°C1 +273)	K2 (°C2 +273)	$(K1/K2)^4$	Relative Radiation
700	20	973	293	122	1.0
900	20	1173	293	255	2.1
1100	20	1373	293	482	3.96
1300	20	1573	293	830	6.83
1500	20	1773	293	1340	11.02
1700	20	1973	293	2056	16.91

In practical terms this means the radiation losses from an open furnace door at 1500°C are 11 times greater than the same furnace at 700°C. A good incentive for the iron and steel melters is to keep the furnace lid closed at all times and maintaining a continuous feed of cold charge onto the molten bath.

Furnace Utilization Factor

Utilization has a critical effect on furnace efficiency and is a factor that is often ignored or under-estimated. If the furnace is at temperature then standby losses of a furnace occur whether or not a product is in the furnace.

Standby Losses

Energy is lost from the charge or its enclosure in the way of heat: (a) conduction, (b) convection; or/ and (c) radiation

Furnace Draft Control

Furnace pressure control has a major effect on fuel fired furnace efficiency. Running a furnace at a slight positive pressure reduces air ingress and can increase the efficiency.

Scale

Scale is the flaky material formed on the surface due to oxidation of iron/steel at high temperature in the presence of excess air. This results in loss of material and hence useful output.

2.9 Data Tables

Table – I: Co-efficient based on the profile of furnace openings (φ)

Shape of opening	Diameter (or) the shortest side ÷ wall thickness							
	0.01	**0.1**	**0.2**	**0.5**	**1.0**	**2.0**	**4.0**	**6.0**
Circular	0.02	0.10	0.18	0.35	0.52	0.67	0.80	0.86
Square	0.02	0.11	0.20	0.36	0.53	0.69	0.82	0.87
Rectangular (2:1)	0.03	0.13	0.24	0.43	0.60	0.75	0.86	0.90
Very long and very narrow	0.05	0.22	0.34	0.54	0.68	0.81	0.89	0.92

Table – II: Specific heat of gases at constant pressure

Temperature (°C)	Specific heat in kcal/Nm^3 °C						
	N_2	CO_2	O_2	H_2O	H_2	SO_2	Air
0	0.311	0.388	0.312	0.341	0.306	0.425	0.311
100	0.311	0.413	0.316	0.344	0.307	0.446	0.311
200	0.312	0.433	0.321	0.349	0.309	0.465	0.312
300	0.313	0.451	0.325	0.352	0.310	0.482	0.316

400	0.317	0.467	0.330	0.357	0.311	0.495	0.318
500	0.319	0.481	0.334	0.364	0.311	0.508	0.322
600	0.322	0.494	0.339	0.370	0.312	0.519	0.324
700	0.325	0.505	0.344	0.376	0.313	0.528	0.328
800	0.329	0.516	0.348	0.382	0.314	0.536	0.332
900	0.332	0.524	0.351	0.388	0.317	0.543	0.334
1000	0.334	0.533	0.355	0.394	0.317	0.549	0.339
1100	0.339	0.541	0.356	0.400	0.319	0.555	0.340
1200	0.340	0.548	0.360	0.407	0.322	0.560	0.344
1300	0.343	0.554	0.362	0.411	0.323	0.564	0.345
1400	0.345	0.560	0.365	0.419	0.325	0.568	0.349
1500	0.348	0.566	0.367	0.424	0.327	0.571	0.350
1600	0.350	0.571	0.368	0.429	0.328	0.574	0.354
1700	0.351	0.576	0.371	0.434	0.330	0.578	0.355
1800	0.354	0.580	0.372	0.440	0.333	0.580	0.356
1900	0.355	0.584	0.375	0.444	0.334	0.582	0.359
2000	0.356	0.588	0.377	0.449	0.337	0.584	0.360

Calculation of mean specific heat

Composition of dry flue gas	**Mass of gas/ kg of fuel**	**Mass fraction**	**Specific heat (kcal/Nm^3 ^{o}C from table @ 655 ^{o}C)**	**Partial specific heat (kcal/Nm^3 ^{o}C) @ 655 ^{o}C**	**Partial specific heat (kcal/kg ^{o}C) @ 655 ^{o}C**
CO2	3.15	0.096	0.4995	0.0479	0.0244
SO2	0.01	0.0003	0.5235	0.0002	0.0001
O2	4.33	0.132	0.3415	0.0450	0.0315
N2	25.37	0.772	0.3235	0.2498	0.1998
	32.86				0.26

Specific heat of dry flue gas at 655 ^{o}C = 0.26 kcal/kg ^{o}C
Specific heat of dry flue gas at 40 ^{o}C = 0.24 kcal/kg ^{o}C
(Calculated in the same way as for 655 ^{o}C)

$$Mean\ specific\ heat\ of\ dry\ flue\ gas, kCal/kg\ ^oC = \frac{(0.26\times655)-(0.24\times40)}{655-40}$$

Mean specific heat of dry flue gas ($C_{p\,fg}$), kCal / kg ^{o}C = 0.26

Sample calculation for CO_2 in the above table

$$Mass\ of\ CO_2\ per\ kg\ of\ fuel\ firing = \frac{\%\ Carbon\ in\ fuel \times Molecular\ weight\ of\ CO_2}{100 \times Molecular\ weight\ of\ Carbon}$$

$$\text{Mass of } CO_2 \text{ per kg of fuel, kg /kg} = \frac{85.9 \times 44}{100 \times 12}$$

$$\text{Mass of } CO_2 \text{ per kg of fuel, kg /kg} = 3.15$$

$$Mass\ fraction\ of\ CO_2\ in\ dry\ flue\ gas = \frac{Mass\ of\ CO_2\ /\ kg\ of\ fuel}{Mass\ of\ dry\ flue\ gas\ /\ kg\ of\ fuel}$$

$$\text{Mass fraction of } CO_2 \text{ in dry flue gas} = \frac{3.15}{32.86}$$

$$\text{Mass fraction of } CO_2 \text{ in dry flue gas} = 0.096$$

$$Partial\ specific\ heat, kCal/kg\ ^oC = \frac{Mass\ fraction \times Specific\ heat\ x\ 22.4}{Molecular\ weight}$$

$$\text{Partial specific heat, kCal/kg } ^oC = \frac{0.096 \times 0.4995 \text{ x } 22.4}{44}$$

$$\text{Partial specific heat, kCal/kg } ^oC = 0.0244$$

$$Specific\ heat\ of\ dry\ flue\ gas,\ kCal/kg\ ^oC = \sum Partial\ specific\ heat$$

$$\text{Specific heat of dry flue gas, kCal/kg } ^oC = 0.26$$

<u>Solved Example</u>:

In oil fired furnace following are the operating parameters:

Capacity of furnace	- 10 T/hr
Daily production operating at 10 hours a day	- 100 T/day
Specific fuel consumption	- 65 litres /T of finished product
Flue gas temperature at the exit of furnace	- 600 ^oC
Ambient temperature	- 30 ^oC
G.C.V of oil	- 10,000 kcal/kg
Theoretical air required for combustion	- 14 kg of air/ kg of fuel
Specific heat of flue gas	- 0.26 kcal/kg^oC
Specific heat of air	- 0.24 kcal/kg^oC
Oxygen in flue gas	- 8 %

The management is planning to install a recuperator to preheat the combustion air upto 200^oC.

Yield without the recuperator	- 90%
Yield after installing the recuperator	- 95%

Calculate:

(i) the percentage heat reduction in flue gas after installation of recuperator
(ii) the increase in daily production due to yield improvement
(iii) specific fuel consumption after installing the heat recovery recuperator (assuming 1 % fuel saving for every 20^{o}C rise in combustion air temperature)

Solution:

(i)

% excess air supplied	= 8/ (21-8)
	= 61.5 %
Actual mass of air supplied	= [1 + (EA/100)] x theoretical air
	= [1 + (61.5/100)] x 14
	= 22.61 kg of air/kg of fuel
Daily fuel consumption	= 65 x 100 = 6500 kg/day
Heat in flue gas	= {6500 + (6500 x 22.61)}x0.26 x(600-30)
	= 22743513 kcal/day
Heat in preheated combustion air	= 6500 x 22.61 x 0.24 x (200 - 30)
	= 5996172 kcal/day
Percentage heat reduction in flue gas	= (5996172 / 22743513)
	= 26.4 %

(ii)

Daily additional production due to yield improvement

	= 100 x 95/90
	= 105.5 Tonnes/day
Additional production	= 105.5 – 100
	= 5.5 T/day

(iii)

Reduction in fuel consumption	= (200 - 30)/20 = 8.5 %
Fuel consumption after waste heat recovery	= 6500 – (6500 x 0.085)
	= 5947.5
Specific fuel consumption	
Daily oil consumption	= 5947.5 kg/day
Production	= 105.5 T/day
Specific fuel consumption	= 5947.5/ 105.5
	= 56.37 kg/tonne

QUESTIONS	
S-1	Give two examples of continuous reheating furnaces?
S-2	Give two examples of batch type reheating furnaces?
S-3	What property of steel is required in evaluating the efficiency of a heat treatment furnace by direct method?
S-4	What are the disadvantages of heating the charge above the optimum temperature in steel re-rolling furnaces?
S-5	Why does radiation heat loss increase considerably with increase in the temperature of a furnace wall?
S-6	Between efficiency and specific energy consumption, which is a better measure of comparing furnaces?
S-7	How will you measure the temperature of the stock inside the furnace?
S-8	What do you mean by scale losses? How do they occur?
S-9	Write down the term which represents the quantity of heat release by natural convection in the equation for determination of heat loss from the surface of a furnace body
S-10	List down two heat losses taking place in oil-fired furnace.
L-1	An air preheater was installed to preheat the combustion air temperature in a steel reheating furnace. The measurements were taken before and after preheater installation. Find out pay back period from the data given below. What is the annual fuel oil bill after heat recovery? Fuel oil consumption = 600 litres/hr Specific gravity = 0.90 Production rate = 9 Ton/hr Furnace operating temperature = 1250^{o}C % O_2 before waste heat recovery = 7.5% % O_2 after waste heat recovery = 5.5% Investment for heat recovery = Rs. 20.00 lakhs Flue gas temperature before heat recovery = 950^{o}C Flue gas temperature after heat recovery = 750^{o}C Ambient temperature = 30^{o}C Fuel oil cost = Rs.25/kg Ambient temperature = 30^{o}C Mean specific heat of flue gas = 0.25 kcal/kg ^{0}C
L-2	The specific heat of dry flue gas at 500 ^{0}C is 0.25 kcal/kg ^{0}C and at 30 ^{0}C is 0.23 kcal/kg ^{0}C. Calculate the mean specific heat.
N-1	Calculate the annual financial loss from an opening in a reheating furnace which is 1m in diameter and having 500 mm wall thickness. The furnace radiates at 1200°C, and is operated for 12 hours a day, 200 days per year. Fuel oil costs are Rs.25,000/- per tonne and the oil has a calorific value of 10,000 kcal/kg. Ambient temperature is 30 ^{0}C.

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Chapter 3

Energy Performance Assessment of Cogeneration Systems

Learning Objectives

In this chapter you will learn about

- Cogeneration performance: terms and definitions
- Turbine stage (isentropic) efficiency calculation
- How to estimate the overall plant heat rate and energy utilization factor?

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Compute steam turbine efficiency**
- ✓ **Estimate the overall plant heat rate**
- ✓ **Estimate the Energy Utilization Factor (EUF) for the cogeneration plant**
- ✓ **Evaluate the efficiency of Heat Recovery Steam Generator (HRSG) in a gas turbine cogeneration plant**

3. ENERGY PERFORMANCE ASSESSMENT OF COGENERATION SYSTEMS WITH STEAM AND GAS TURBINES

3.1 Introduction

Cogeneration systems can be broadly classified as those using steam turbines, Gas turbines and DG sets. Steam turbine cogeneration systems involve different types of configurations with respect to mode of power generation such as extraction, back pressure or a combination of backpressure, extraction and condensing.

Gas turbine with heat recovery steam generators is another mode of cogeneration. Depending on power and steam load variations in the plant the entire system is dynamic. A performance assessment would yield valuable insights into cogeneration system performance and need for further optimization.

3.2 Purpose of the Performance Test

The purpose of the cogeneration plant performance test is to determine the power output and plant heat rate. In certain cases, the efficiency of individual components like steam turbine is addressed specifically where performance deterioration is suspected. In general, the plant performance will be compared with the base line values arrived at for the plant operating condition rather than the design values. The other purpose of the performance test is to show the maintenance accomplishment after a major overhaul. In some cases the purpose of evaluation could even be for a total plant revamp.

3.3 Performance Terms and Definitions

Overall Plant Performance

$$Overall\ plant\ fuel\ rate, kg/kWh\,(or)\,Sm^3/kWh = \frac{Fuel\ consumption, kg/hr\ or\ Sm^3/hr}{Power\ output, kW}$$

$$Energy\ Utilisation\ Factor,\ EUF = \frac{Electrical\ output, Q_e + Thermal\ ouput, Q_{th}}{Fuel\ heat\ input, Q_f}$$

Steam Turbine Performance

$$\text{Turbine stage (isentropic) efficiency, \%} = \frac{\text{Actual enthalpy drop across the turbine, kCal/kg}}{\text{Stage (Isentropic) enthalpy drop across the turbine, kCal/kg}} \times 100$$

$$\text{Overall plant heat rate, kCal/kWh} = \text{Overall plant fuel rate, kg / kWh} \times \text{GCV of fuel, kCal / kg}$$

Gas Turbine and Heat Recovery Steam Generator Performance

$$\text{Overall plant heat rate, kCal/kWh} = \text{Overall plant fuel rate, } Sm^3 \text{ / kWh} \times \text{NCV of gas, kCal / } Sm^3$$

$$\eta_{HRSG} = \frac{\text{Steam flow rate} \times (\text{Enthalpy of steam} - \text{Enthalpy of feed water}) \times 100}{(\text{Exhaust gas flow rate x I/L enthalpy}) + (\text{Auxiliary fuel} \times \text{GCV})}$$

3.4 Reference Standards

ASME performance test codes

3.5 Field Testing Procedure

The test procedure for each cogeneration plant will be developed individually taking into consideration the plant configuration, instrumentation and plant operating conditions.

3.5.1 Test Duration

The test duration is site specific and in a continuous process industry, 8-hour test data should give reasonably reliable data. In case of an industry with fluctuating electrical/steam load profile, a set of 24-hour data sampling to be taken for a representative period.

3.5.2 Measurements and Data Collection

The suggested instrumentation (online/ field instruments) for the performance measurement is as under:

Steam flow measurement	:	Orifice flow meters
Fuel flow measurements	:	Volumetric measurements / Mass flow meters
Air flow / Flue gas flow	:	Venturi / Orifice flow meter / Ion gun / Pitot tubes
Flue gas analysis	:	Zirconium Probe Oxygen analyser
Unburnt analysis	:	Gravimetric analysis
Temperature	:	Thermocouple

Cooling water flow : Orifice flow meter / weir /channel flow/ non-contact flow meters
Pressure : Bourdon Pressure Gauges
Power : Trivector meter / Energy meter
Condensate : Orifice flow meter

It is essential to ensure that the data is collected during steady state plant running conditions. Among others the following are essential details to be collected for cogeneration plant performance evaluation.

I. Thermal Energy:

S.No.	Description	Flow	Pressure	Temperature
1	Steam inlet to turbine	✓	✓	✓
2	Fuel input to Boiler /Gas turbine	✓	-	-
3	Combustion air	✓	✓	✓
4	Extraction steam to process	✓	✓	✓
5	Back Pressure Steam to Process	✓	✓	✓
6	Condensing steam	✓	✓	✓
7	Condensate from turbine	✓	-	✓
8	Turbine bypass steam	✓	-	-
9	Flue gas to HRSG	-	✓	✓
10	Exit flue gas	-	-	✓+ composition
11	Cooling water to condenser	✓	✓	✓

II. Electrical Energy:

1. Total power generation for the trial period from individual turbines.
2. Hourly average power generation
3. Quantity of power import from utility (Grid)
4. Auxiliaries power consumption

3.5.3 Calculations for Steam Turbine Cogeneration System

The process flow diagram for cogeneration plant is shown in Figure 3.1. The following calculation procedures have been provided in this section.

- Turbine cylinder efficiency
- Overall plant heat rate

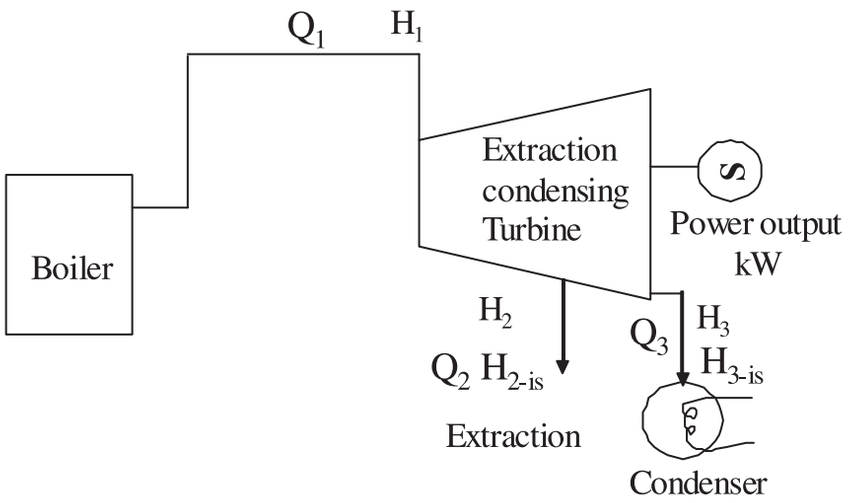


Figure 3.1 Process Flow Diagram for Cogeneration Plant

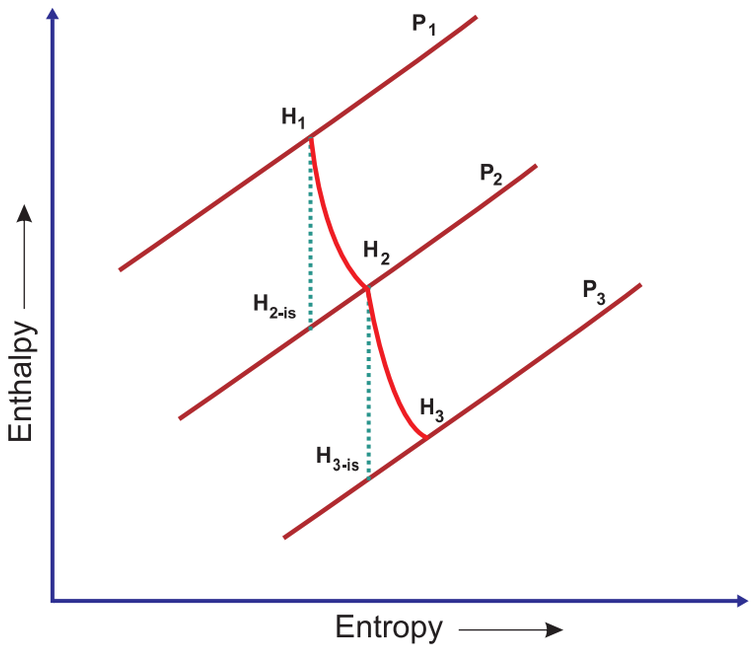


Figure 3.2 Mollier diagram

Step 1:

Calculate the **actual** heat extraction in turbine at each stage,

Steam Enthalpy at turbine inlet : H_1 kcal / kg

Steam Enthalpy at 1st extraction : H_2 kcal / kg

Steam Enthalpy at Condenser : H_3^* kcal / kg

* Due to wetness of steam in the condensing stage, the enthalpy of steam cannot be considered as equivalent to saturated steam. Typical dryness value is 0.88 – 0.92. This dryness value can be used as first approximation to estimate heat drop in the last stage. However it is suggested to calculate the last stage efficiency from the overall turbine efficiency and other stage efficiencies.

Heat extraction from inlet to extraction : $H_1 - H_2$ kCal / kg

Heat extraction from Extraction to condenser : $H_2 – H_3$ kCal / kg

Step 2:

From Mollier diagram (H-f Diagram) estimate the theoretical heat extraction for the conditions mentioned in Step 1. Towards this:

a) Plot the turbine inlet condition point (H_1) in the Mollier chart – corresponding to steam pressure (P_1) and temperature.

b) Since expansion in turbine is an adiabatic process, the entropy is constant. Hence draw a vertical line from inlet point (parallel to y-axis) upto the extraction pressure (P_2). Read the corresponding enthalpy $H_{2\text{-}is}$.

c) Plot the extraction condition point (H_2) in the Mollier chart – corresponding to steam pressure (P_2) and temperature.

d) Draw a vertical line from extraction point (parallel to y-axis) upto the condensing pressure (P_3). Read the corresponding enthalpy $H_{3\text{-}is}$.

e) Compute the theoretical heat drop for different stages of expansion.

Step 3:

Compute turbine stage (isentropic) efficiency

$$Efficiency\ of\ extraction\ \ stage = \frac{Heat\ extraction\ actual}{Heat\ extraction\ theoretical} = \frac{H_1 - H_2}{H_1 - H_{2-is}}$$

$$Efficiency\ of\ conden\sin g\ \ stage = \frac{Heat\ extraction\ actual}{Heat\ extraction\ theoretical} = \frac{H_2 - H_3}{H_2 - H_{3-is}}$$

Step 4:

To calculate the turbine power output (P_t)

$$Turbine\ power\ output\ (P_t), MW = \frac{Q_1 \times (H_1 - H_2) + Q_3 \times (H_2 - H_3)}{860 \times 1000}$$

Where,

Q_1, Q_2 & Q_3 - Steam flow rate in kg/hr

H - Enthalpy in kcal/kg

Step 5:

To calculate the generator power output (P_g)

$$Generator\ power\ output\ (P_g), MW = P_t \times \eta_{\ m} \times \eta_{\ gear} \times \eta_{\ gen}$$

Where,

$\eta_{\ m}$ - Mechanical efficiency of the turbine in %

$\eta_{\ gear}$ - Efficiency of the gear box in %

$\eta_{\ gen}$ - Efficiency of the generator in %

3.6 Examples

Example 3.6.1

From the data given for an extraction condensing turbine, calculate the stage-wise (isentropic) turbine efficiency and power output.

PARAMETERS	UNITS	Data
Main Steam		
Steam Pr.	kg/cm^2(g)	68
Steam Temp.	^{o}C	485
Steam Flow	TPH	19.2
Enthalpy	kcal/kg	807.9
1ST Extraction		
Steam Pr.	kg/cm^2(g)	9
Steam Temp.	^{o}C	280
Steam Flow	TPH	8
Enthalpy (Actual)	kcal/kg	719.9
Enthalpy (isen)	kcal/kg	684.2

Condenser		
Cond. Vac.	kg/cm^2(g)	-0.9
Temp.	^{o}C	45.4
Cond. Flow	TPH	11.2
Dryness	%	90
Enthalpy	kcal/kg	561
Enthalpy (isen)	kcal/kg	534.2

Calculation

$$Turbine\ stage\ (isentropic)\ efficiency, \% = \frac{Actual\ enthalpy\ drop\ across\ the\ turbine, kCal/kg}{Stage\ (Isentropic)\ enthalpy\ drop\ across\ the\ turbine, kCal/kg}\ x\ 100$$

Extraction stage (isentropic) efficiency = (807.9 – 719.9) / (807.9 – 684.2)
= 71.14 %

Condensing stage (isentropic) efficiency = (719.9– 561) / (719.9 – 534.2)
= 85.57 %

Extraction stage power output = 19200 x (807.9 – 719.9)/860
= 1944 kW

Condensing stage power output = 11200 x (719.9 – 561)/860
= 2069.4 kW

Total power output = 4013 kW

Example 3.6.2

Calculate the following performance parameters of the gas turbine, details of which are given below

1. Overall plant fuel rate
2. Overall plant heat rate
3. Thermal efficiency of HRSG
4. Energy Utilisation Factor (EUF)

Description		
Parameter	**Unit**	**Quantity**
Test duration	hours	4
Gas turbine data		

Gas turbine compressor inlet conditions		
Air temperature	^{0}C	37.0
Pressure	kg/cm^2 (g)	1.0332
Dry bulb temperature	^{0}C	36.5
Wet bulb temperature	^{0}C	28.0
Diff. Pressure - Inlet air filter	mmWC	35.8

Fuel Data		
Fuel fired	Natural gas	
Fuel flow rate	sm^3/hour	1312
Lower heating value of NG	kcal/sm^3	9465
Auxiliary fuel for HRSG		NIL
Exhaust flue gas conditions		
Flow	kg/sec	14.32
Temperature	^{0}C	548
Specific heat of flue gas	kcal/kg ^{o}C	0.25
Generator data		
Average power output	kW	3994.5
Power factor		0.875
Waste heat recovery boiler data		
Exhaust gas temp at inlet	^{0}C	542
Exhaust gas temp exit boiler	^{0}C	131.4
Steam parameters at WHRB exit		
Flow	MT/hour	9.145
Temperature	^{0}C	195.5
Pressure	kg/cm^2 (g)	8.05
Feed water inlet parameters		
Flow	kg/hour	9605
Temperature at drum inlet	^{0}C	105
Pressure	kg/cm^2 (g)	12.4
Enthalpy at drum inlet	kcal/kg	105

1. Determination of overall plant fuel rate

Fuel consumption = 1312 Sm^3/hr
Electrical power output = 3994.5 kW
Overall plant fuel rate = 1312/3994.5
= 0.32844 Sm^3/kWh

2. Overall plant heat rate

$$Overall\ \ plant\ heat\ rate, kCal/kWh$$
$$= Overall\ \ plant\ \ fuel\ rate,\ Sm^3\ /\ kWh \times GCV\ of\ \ fuel,\ kCal\ /\ Sm^3$$
$$= 0.32844\ \ \times 9465$$
$$= 3109\ \text{kCal/kWh}$$

3. Thermal efficiency calculations for HRSG

$$\eta_{HRSG} = \frac{Steam\ \ flow\ rate \times (enthalpy\ of\ \ steam - enthalpy\ of\ \ feed\ water)\ x\ 100}{(Exhaust\ gas\ flow\ rate\ \ x\ \ I/L\ enthalpy) + (Auxiliary\ \ fuel \times GCV)}$$

$$\eta_{HRSG} = \frac{9145\ x\ (678.4 - 105)\ x\ 100}{51561\ x\ 0.25\ x\ 542}$$

$$= 75\ \%$$

* No Auxiliary fuel used in this case.

4. Energy Utilisation Factor (EUF)

$$EUF = \frac{P_e + Q_{th}}{Q_f}$$

$$EUF = \frac{(3994.5 \times 860) + \{9145 \times (678.4 - 105)\}}{1311.97 \times 9465}$$

$$= 70\ \%$$

3.7 Case Study of Bottoming Cycle Cogeneration in a Cement Industry

Waste heat sources in a cement plant

In cement manufacturing process, raw materials are burnt in the Rotary Kiln and the fuel used for combustion generates huge quantity of exhaust gases. A part of the heat in the flue gas is utilized for pre heating the raw materials going to kiln in preheaters. Along with this the heat in flue gas is also used remove the moisture of coal (used as fuel) and limestone (main raw material for cement production) during grinding. Further the heat may also be used to dry Puzzolonic materials such as fly ash or slag used in the manufacture of blended cement.

Based on the number of preheater stages, kiln gases exit at around 300 – 400°C in case of 4 stage pre-heater and 200 – 300°C in case of 5 – 6 stage pre-heater. The quantity of heat from pre-heater exit gases ranges from 180–250 kcal/kg clinker.

The solid material i.e. clinker coming out of the rotary kiln is at around 1000 °C and is cooled to 100-120 °C temperature using ambient air in clinker cooler. This generates hot air of about 200-300 °C having heat of 80-130 kcal/kg clinker. Part of the hot air generated is used as combustion air in kiln furnaces & remaining is exhausted to atmosphere.

Waste gas discharged from Kiln Preheater and clinker cooler thus contains useful energy that can be converted into power by installation of waste heat boiler that runs a steam turbine. The generation potential depends on the Kiln capacity, number of pre-heater stages, heat required to remove moisture in raw material and coal.

Waste heat recovery based power generation

The waste heat recovery (WHR) system, effectively utilises the available waste heat from exit gases of pre-heater and clinker cooler. The WHR system consists of **Suspension pre-heater (SP) boiler**, **Air Quenching Chamber (AQC) boiler**, steam turbine generator, distributed control system (DCS), water-circulation, system and **dust-removal system** etc as shown in Figure 3.3

Process requirements decide the output temperature of flue gas from the waste heat recovery boilers thus deciding available heat to be recovered. Seasonal variations in the demand of flue gas for drying raw material also to be met while designing the system.

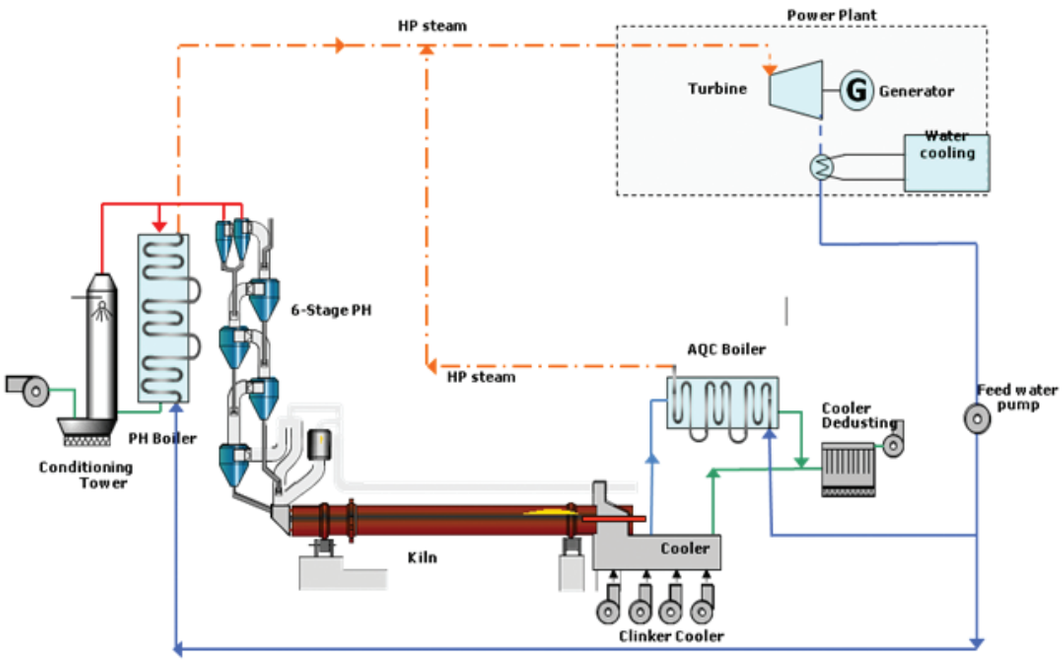


Figure 3.3 Schematic of steam Rankine cycle based waste heat recovery power generation

Calculation of power generation potential

1. Kiln capacity: 4300 Tons per Day (179.167 Tons Per Hour)

2. No of stages in the preheater: 5

3. Preheater exit gas details:

Volume (m_{PH}): 167559 Nm^3/hr
Specific heat capacity (C_{PH}) : 0.355 kcal/kg/ Deg C
Inlet Temperature T: 295 Deg C
Outlet temperature with WHRB: 195 Deg C

4. Cooler exit gas details:

Volume (m_C) : 91000 Nm^3/hr
Specific heat capacity (C_C) : 0.316 kcal/kg/ Deg C
Inlet Temperature T: 360 Deg C
Outlet temperature with WHRB: 130 Deg C

5. Overall Conversion efficiency –Pre heater section: 20%

6. Overall Conversion efficiency- Cooler section: 21 %

Calculations:

Heat recoverable in Preheater gas $= m_{PH} * C_{PH} * \Delta T$
$= 167559 \times 0.316 \times (295-195)$
$= 5.948$ million kcal/hr

Total power generation possible considering 20 % efficiency in Preheater circuit
$= 1.1896$ million kcal/hr

Considering 860 kcal/kWh, the total power generation possible $= 1.1896 \times 10^6/860$
$= 1.383$ MW

Heat recoverable in Cooler gas $= m_C * C_C * \Delta T$
$= 91000 \times 0.316 \times (360-130)$
$= 6.613$ million kcal/hr

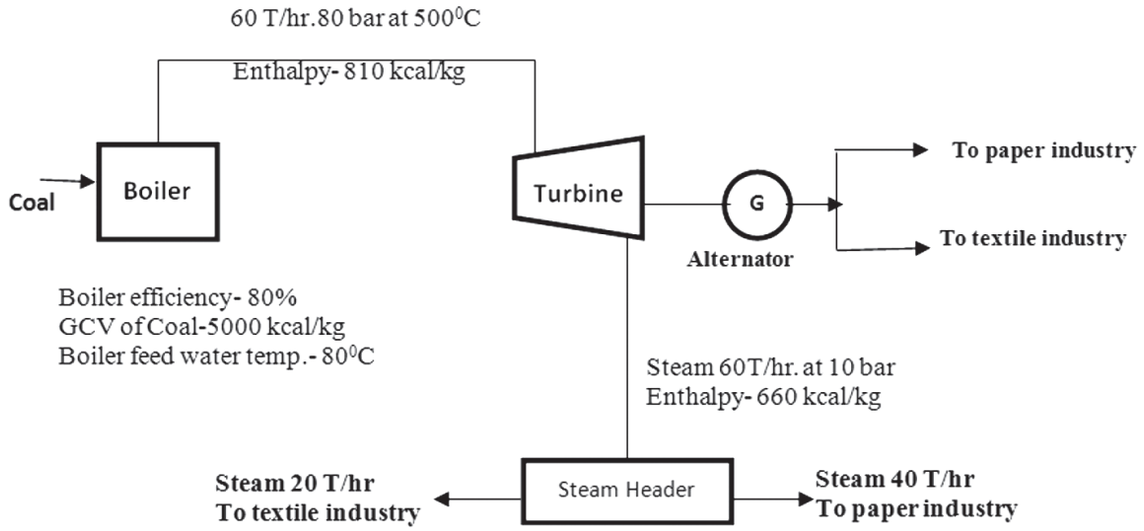
Total power generation possible considering 21 % efficiency in Cooler circuit
$= 1.389$ million kcal/hr

Considering 860 kcal/kWh, the total power generation possible $= 1.389 \times 10^6/860$
$= 1.615$ MW

Total power generation possible = 1.383 + 1.615 = 3.0 MW

Solved Example:

A common plant facility is installed to provide steam and power to textile and paper plant with a co-generation system. The details and operating parameters are given below:



Other data:

- Turbine, alternator and other losses = 8%
- Specific steam consumption in paper industry= 5 Tons/Ton of paper
- Specific power consumption in paper industry= 600 kWh/Ton of paper

Calculate:

i. Coal consumption in boiler per hour or per day.

ii. Power generation from co-generation plant

iii. If 10% is auxiliary power consumption in co-generation plant, how much power is consumed by the textile industry per hour?

iv. What is the gross heat rate of turbine?

Solution:

i) Boiler efficiency $= \frac{\text{Steam production (Steam enthalpy- Feed water enthalpy)}}{\text{Quantity of coal x G.C.V. of coal}}$

Quantity of coal = 60,000 (810-80)/ 0.8 x 5000

= 10.95 tons/hr.

ii) Gross power generation from co-generation plant

Total enthalpy input to turbine = 60,000 x 810 = 48.6 Million kcal.

Total enthalpy out put through back pressure = 60,000* 660 = 39.6 Million kcal

Enthalpy difference = 48.6- 39.6 = 9 Million kcal/hr

Turbine, alternator and other losses (8%) = 9x0.08 = 0.72 Million kcal/hr

Useful energy for power generation = 9 - 0.72 = 8.28 Million kcal/hr

Power generation from co-generation plant = 8.28 x 10^6/860 = **9628 kWh**

iii) If 10% is auxiliary power consumption in co-generation plant, power consumed by textile industry

10% of total power generation = 9628 x 0.10 = 962.8 kWh

Total power consumed by industries = 9628 – 962.8 = 8665.2 kWh

Total steam consumption in paper plant 40 tons/hr. and specific steam consumption 5 ton/ton of paper. So Paper production per hour is 8 tons.

Specific power consumption = 600kWh/ton.

Total power consumption in paper industry = 8 x 600 = 4800kWh

Total power consumption in textile industry = 8665.2- 4800 = **3865.2 kWh**

iv) Gross heat rate = Input enthalpy – Output enthalpy/ Gross generation

= (48.6- 39.6) 10^6/ 9628 = **934.7 kcal/kWh**

QUESTIONS	
S-1	What is meant by overall plant heat rate? What is its significance?
S-2	Draw the schematic diagram of topping cycle cogeneration system.
S-3	Between back pressure turbine and condensing turbine which will have more power generation efficiency? Why?
S-4	What is meant by turbine stage (isentropic) efficiency? How is it determined?
S-5	Which parameter is assumed to remain constant in an ideal expansion through steam turbine?
S-6	Starting from steam input to the turbine what are the various efficiencies considered in determining the electrical power output from a steam turbine based cogeneration plant?
S-7	Between a thermal power plant and a cogeneration plant with a back pressure turbine, which will have a higher heat rate? Why?
S-8	An industry with cogeneration system requires process steam at 12 bar and 3 bar. What type of turbine is most suitable for this application?
S-9	A cogeneration plant with a back pressure turbine has a constant steam demand and fluctuating power demand. What options are available to meet the fluctuating power demand?
S-10	Other than exhaust gas what is the major source of waste heat recovery in a water cooled DG set?
L-1	The parameters for back pressure steam turbine cogeneration plant is given below Inlet Steam: P =16 kg/cm^2 , T = 310^{o}C , Q = 9000kg/hr Outlet Steam: P = 5.0 kg/cm^2, T = 235^{o}C , Q = 9000kg/hr Find out the turbine Stage (isentropic) efficiency?
L-2	Explain why EUF is a better indicator of cogeneration performance than heat rate.
N-1	A process plant is installing a 5 MW gas turbine cogeneration system with 12 TPH waste heat boiler to meet the power and steam demand of the plant. The plant will operate at 90% of capacity, meeting the entire power requirement of the plant, which is presently drawn from grid supply. The co-gen plant will also meet the steam requirement of 10 TPH, which is presently generated in a gas fired boiler with 86% efficiency on N.C.V. basis. Calculate the differential cost between cogenerated power and grid power per unit and also the additional natural gas requirement per day based on the following data. Capacity of gas turbine = 5000 kW Plant load factor = 90% Auxiliary power consumption = 1% Operating hrs. per annum = 8000 Net calorific value of natural gas = 9500 kcal / Sm^3 Cost of natural gas = Rs.8 / Sm^3 Steam produced by co-gen waste heat = 10 TPH boiler

	Annual expenditure towards depreciation and interest	=	Rs 500 lacs
	Annual expenditure for operation & maintenance of co-gen plant	=	Rs 200 lacs
	Heat Rate of gas turbine on NCV	=	3050 kcal / kWh
	Cost of electric power from grid supply	=	Rs. 4.5/ kWh
	Enthalpy of steam	=	665 kcal/ kg
	Feed water temperature	=	85^{o}C

REFERENCES

1. NPC report on 'Assessing cogeneration potential in Indian Industries'
2. Energy Cogeneration Handbook, George Polimeros, Industrial Press Inc.
3. Inputs from ACC Ltd.

Chapter 4

Energy Performance Assessment of Heat Exchangers

Learning Objectives

In this chapter you will learn about

- Heat exchangers performance terms
- Nomenclature of heat exchangers
- Methodology of heat exchanger performance assessment
- Determination of overall heat transfer coefficient and effectiveness

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Perform a test to determine the overall heat transfer coefficient of heat exchangers in the field**
- ✓ **Determine the effectiveness of heat exchangers in the field**

4. ENERGY PERFORMANCE ASSESSMENT OF HEAT EXCHANGERS

4.1 Introduction

Heat exchangers are equipment that transfers heat from one medium to another. Shell and tube heat exchangers are used extensively through out the process and power industry and as such a basic understanding of their design, construction and performance is important to the practicing engineer. The proper design, operation and maintenance of heat exchangers will make the process energy efficient and minimize energy losses.

4.2 Purpose of the Performance Test

Heat exchanger performance can deteriorate with time, off design operations and other interferences such as fouling, scaling, corrosion etc. It is necessary to assess periodically the heat exchanger performance in order to maintain them at a high efficiency level. This section comprises certain proven techniques of monitoring the performance of heat exchangers, coolers and condensers from observed operating data of the equipment.

The objective of performance assessment is to determine the heat exchanger duty, overall heat transfer coefficient, heat exchanger effectiveness, process / utility side pressure drop. Any deviation from the design will indicate occurrence of fouling.

4.3 Performance Terms and Definitions

Overall heat transfer coefficient, U

The overall heat transfer coefficient, U, represents how easily the heat can move. A smaller value of U indicates the difficulty in the transfer of heat and vice versa. Heat exchanger performance is normally evaluated by the overall heat transfer coefficient *U* that is defined by the equation:

$$Q = U \times A \times LMTD$$

Where

Q = Heat transferred in **kcal/hr**
A = Heat transfer surface area in $\mathbf{m^2}$
LMTD = Log Mean Temperature Difference in ^{0}**C**
U = Overall heat transfer Coefficient **kcal/hr/m^2/^{0}C**

When the hot and cold stream flows and inlet temperatures are constant, the heat transfer coefficient may be evaluated using the above formula.

Heat Exchanger Effectiveness,

$$\varepsilon = \frac{\text{Actual heat transfer rate, kcal/h}}{\text{Maximum possible heat transfer rate, kcal/h}}$$

$$= Q / Qmax = Q / (Cmin \times \Delta Tmax)$$

Where,

C = Heat Capacity, **kcal/h ^{0}C**

= Mass flow rate (kg/hr) x Specific heat capacity of the fluid (kcal/kg^{0}C)

Cmin = Lower of the two fluids heat capacities, **kcal/h ^{0}C**

ΔTmax = Maximum temperature difference from the terminal stream temperatures, **^{0}C**

The most commonly used methods for heat transfer analysis are LMTD-F method and the Effectiveness–NTU method. This section gives an overview of LMTD-F method.

LMTD

The LMTD is Logarithmic Mean Temperature Difference, used to determine the temperature driving force for heat transfer in heat exchangers. It is determined by the relationship of the fluid temperature differences at the terminals of the heat exchanger.

The LMTD Correction Factor, F

If the flow is true counter current, the LMTD calculated is used directly in the basic heat transfer equation. If the flow is not true counter current (i.e., more tube passes than shell passes), LMTD must be corrected and also to account for cross flow.

In multi-pass shell-and-tube exchangers, the flow pattern is a mixture of co-current and counter current flow, as the two streams flow through the exchanger in the same direction on some passes and in the opposite direction on others. For this reason, the mean temperature difference is not equal to the logarithmic mean. However, it is convenient to retain the LMTD by introducing a Correction Factor, F, which is appropriately termed as the LMTD correction factor.

Fouling Factor

One of the important heat-exchanger parameters related to surface conditions is termed as the fouling factor. The fouling factors to be used in the design of heat exchangers are normally specified by the client, based on his experience of running his plant or process to simulate dirt accumulation on the heat transfer surfaces, but if these are not restricted to proper levels, they can totally negate any benefits

generated by skillful design. The fouling factor represents the theoretical resistance to heat flow due to a build up of a layer of dirt or other fouling substance on the tube surfaces of the heat exchanger but they are often overstated by the end user in an attempt to minimize the frequency of cleaning. In reality they can, if badly chosen, lead to increased cleaning frequency. The fouling factor increases with increased fouling and causes a drop in the heat exchanger effectiveness. Common types of fouling are chemical, biological, deposition and corrosion fouling.

4.4 Industrial Heat Exchangers

The most common types of commercially available heat exchangers are the shell-and-tube exchanger.

Nomenclature

A typical heat exchanger is shown in figure 4.1 with nomenclature.

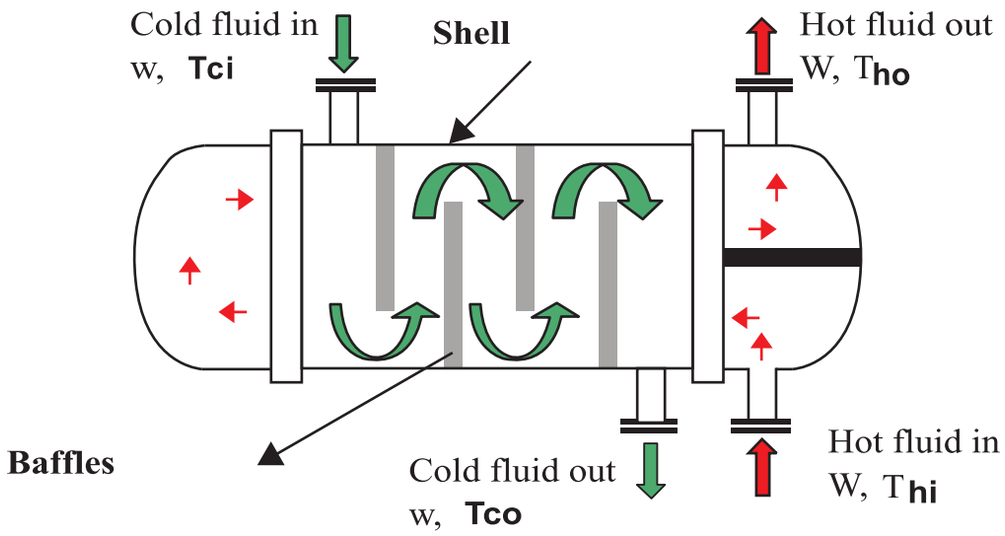


Figure 4.1 Typical Shell and Tube Heat Exchanger

Heat duty of the exchanger can be calculated either on the hot side fluid or cold side fluid as given below.

Heat Duty for Hot fluid, $Q_h = W \times C_{ph} \times (T_{hi}-T_{ho})$ …….. Eqn-1

Heat Duty for Cold fluid, $Q_c = w \times C_{pc} \times (T_{co}-T_{ci})$ ……... Eqn-2

If the operating heat duty is less than design heat duty, it may be due to heat losses, fouling in tubes, reduced flow rate (hot or cold) etc. Hence, for simple performance monitoring of exchanger, heat duty may be considered as factor of performance irrespective of other parameter. A deviation in heat duty may not be a conclusive indicator of fouling due to variation in the process condition. However, the overall heat transfer coefficient offers itself as a reliable indictor of fouling. Hence the performance assessment of the heat exchanger is carried out by determination of overall heat transfer coefficient in the field.

4.5 Methodology of Heat Exchanger Performance Assessment

4.5.1 Procedure for determination of Overall heat transfer Coefficient, U

This is a fairly rigorous method of monitoring the heat exchanger performance by calculating the overall heat transfer coefficient periodically. Technical records are to be maintained for all the exchangers, so that problems associated with reduced efficiency and heat transfer can be identified easily. The record should basically contain historical heat transfer coefficient data versus time / date of observation. A plot of heat transfer coefficient versus time permits rational planning of an exchanger-cleaning program.

The heat transfer coefficient is calculated by the equation

$$U = Q / (A \times LMTD)$$

Where Q is the heat duty, A is the heat transfer area of the exchanger and LMTD is temperature driving force.

The step by step procedure for determination of Overall heat transfer Coefficient is described below.

Step – A

Monitoring and reading of steady state parameters of the heat exchanger under evaluation are tabulated as below:

Parameters	Units	Inlet	Outlet
Hot fluid flow,W	kg/h		
Cold fluid flow,w	kg/h		
Hot fluid Temp, T_h	^{O}C		
Cold fluid Temp,T_c	^{O}C		
Hot fluid Pressure,P	bar g		
Cold fluid Pressure, p	bar g		

Step – B

With the monitored test data, the physical properties of the stream can be tabulated as required for the evaluation of the thermal data

Parameters	Units	Inlet	Outlet
Hot fluid density, ρ_h	kg/m^3		
Cold fluid density, ρ_c	kg/m^3		
Hot fluid Viscosity, μ_h	MpaS*		
Cold fluid Viscosity, μ_c	MPaS		
Hot fluid Thermal Conductivity, k_h	kW/(m. K)		

Cold fluid Thermal Conductivity, k_c	kW/(m. K)		
Hot fluid specific heat Capacity, C_{ph}	kJ/(kg. K)		
Cold fluid specific heat Capacity, C_{pc}	kJ/(kg. K)		

* MpaS – Mega Pascal Second

Density and viscosity can be determined by analysis of the samples taken from the flow stream at the recorded temperature in the plant laboratory. Thermal conductivity and specific heat capacity if not determined from the samples can be collected from handbooks.

Step – C

Calculate the thermal parameters of heat exchanger and compare with the design data

Parameters	Units	Test Data	Design Data
Heat Duty, Q	kW		
Hot fluid side pressure drop, ΔP_h	bar		
Cold fluid side pressure drop, ΔP_c	bar		
Temperature Range hot fluid , ΔT_h	^{O}C		
Temperature Range cold fluid , ΔT_c	^{O}C		
Capacity ratio, R	-----		
Effectiveness, S	-----		
Corrected LMTD, MTD	^{O}C		
Heat Transfer Coefficient, U	kW/(m^2. K)		

Step – D

The following formulae are used for calculating the thermal parameters:

1. **Heat Duty**, $Q = q_s + q_l$

Where,

q_s is the sensible heat and q_l is the latent heat

For Sensible heat

$q_s = W \times C_{ph} \times (T_{hi} - T_{ho}) / 3600$ in kW

(or)

$q_s = w \times C_{pc} \times (T_{co} - T_{ci}) / 3600$ in kW

For Latent heat

$q_l = (W \times \lambda_h) / 3600$, in kW,

where λ_h – Latent heat of Condensation of a hot condensing vapor, kJ/kg

(or)

$q_l = (w \times \lambda_c) / 3600$, in kW, where λ_c - Latent heat of Vaporization

2. Hot Fluid Pressure Drop, $\Delta P_h = P_i - P_o$

3. Cold fluid pressure drop, $\Delta P_c = p_i - p_o$

4. Temperature range hot fluid, $\Delta T_h = T_{hi} - T_{ho}$

5. Temperature range cold fluid, $\Delta T_c = T_{co} - T_{ci}$

6. Calculate LMTD

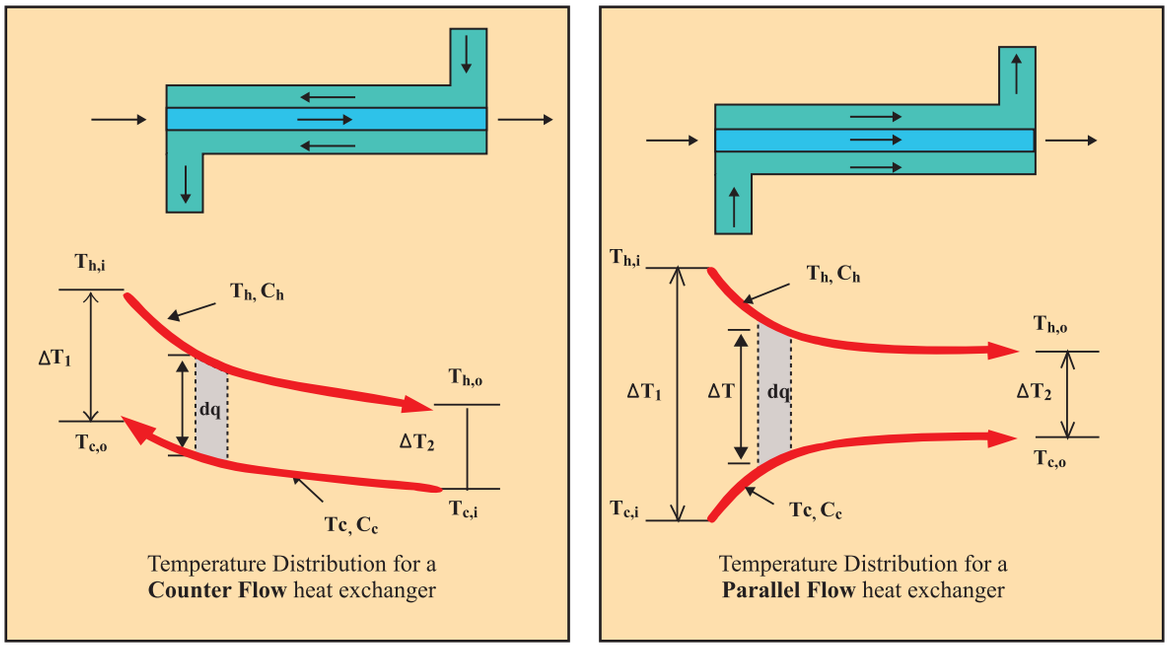


Figure 4.2 Temperature Distributions for a Counter and Co-Current Flow Heat Exchangers

Calculation of LMTD:

LMTD Counter current Flow = $((T_{hi}-T_{co})-(T_{ho}-T_{ci})) / \ln ((T_{hi}-T_{co})/(T_{ho}-T_{ci}))$

LMTD Co current Flow = $((T_{hi}-T_{ci}) -(T_{ho}-T_{co})) / \ln ((T_{hi}-T_{ci})/ (T_{ho}-T_{co}))$

7. The LMTD Correction Factor, F

The LMTD correction factor is a function of the temperature effectiveness and the number of tube and shell passes and is correlated as a function of two dimensionless temperature ratios. Let R and P be the two dimensionless parameters used to calculate LMTD correction factor defined by the equations below.

$$R = \frac{T_a - T_b}{t_b - t_a} \qquad P = \frac{t_b - t_a}{T_a - t_a}$$

Where,

Ta = inlet temperature of shell-side fluid
Tb = outlet temperature of shell-side fluid
t_a = inlet temperature of tube-side fluid
t_b = outlet temperature of tube-side fluid

For R ≠ 1, compute:

$$\alpha = \left[\frac{1 - RP}{1 - P}\right] 1/N$$

$$S = \frac{\alpha - 1}{\alpha - R}$$

$$F = \frac{\sqrt{R^2+1} \quad \ln\left(\frac{1 - S}{1 - RS}\right)}{(R - 1) \ln\left(\frac{2 - S\,(R+1 - \sqrt{R^2+1}\,)}{2 - S\,(R+1 + \sqrt{R^2+1}\,)}\right)}$$

For R = 1, compute:

$$S = \frac{P}{N - (N - 1)\,P} \qquad F = \frac{S\sqrt{2}}{(1 - S) \ln\left(\frac{2 - S\,(2 - \sqrt{2})}{2 - S\,(2 + \sqrt{2})}\right)}$$

Where,

N = Number of shell-side passes
S, α = Parameters used to calculate LMTD correction factor defined by the equations given above.

8. Corrected LMTD = F x LMTD

9. Overall Heat Transfer Co-efficient

$$U = Q / (A \times \text{Corrected LMTD})$$

4.5.2 Heat Exchanger Effectiveness

The heat recovery capability of a heat exchanger is characterized by means of an index referred as the "Heat Exchanger Effectiveness", is a measure of thermal performance.

Calculating the heat exchanger effectiveness helps engineers,

- To predict how a given heat exchanger will perform a new job.
- To predict the stream outlet temperatures without a trial-and-error solution that would otherwise be necessary.

> ***Definition:***
>
> *"The Heat Exchanger Effectiveness is defined for a given heat exchanger of any flow arrangement as the ratio of the actual amount of heat transferred to the maximum possible amount of heat that could be transferred between the two streams with an infinite area".*

The latter is the rate of heat transfer that would occur in a counter-flow exchanger having infinite heat transfer area. In such an exchanger, one of the fluid streams will gain or lose heat until its outlet temperature equals the inlet temperature of the other stream.

The fluid that experiences this maximum temperature change is the one having the smaller value of Heat Capacity, ***C = mass flow rate x specific heat capacity at constant pressure***, as can be seen from the energy balance equations for the two streams.

Thus, if the hot fluid has the lower value of C, we will have $T_{ho} = T_{ci}$, and:

$$Q_{max} = W \times C_{ph} \times (T_{hi}-T_{ci}) = C_{min} \times (T_{hi}-T_{ci})$$

On the other hand, if the cold fluid has the lower value of C, then $T_{co} = T_{hi}$, and:

$$Q_{max} = w \times C_{pc} \times (T_{hi}-T_{ci}) = Cmin \times (T_{hi}-T_{ci})$$

Thus, in either case

$$\mathbf{Q_{max} = C_{min} (T_{hi} - T_{ci}) = C_{min} \times \Delta T_{max}}$$

Where, $\Delta T_{max} = (T_{hi} - T_{ci})$ is the maximum temperature difference from the terminal stream temperatures.

By definition the effectiveness, ε, is given by:

$$\mathbf{\varepsilon = Q / Qmax = Q / (Cmin \times \Delta Tmax)}$$

Heat capacity ratio, r = Cmin / Cmax = (W x Cph) / w x Cpc)

It should be emphasized that the term effectiveness may not be confused with efficiency. The use of the term efficiency is generally restricted to (1) the efficiency of conversion of energy form A to energy form B or (2) a comparison of actual system performance to the ideal system performance, under comparable operating conditions, from energy point of view.

Since we deal here with a component heat exchanger and there is no conversion of different forms of energy in a heat exchanger (although the conversion between heat flow and enthalpy change is present), the term effectiveness is used to designate the efficiency of a heat exchanger. The consequence of the first law of thermodynamics is the energy balance, and hence the definition of the exchanger explicitly uses the first law of thermodynamics.

For air-to-air heat exchangers, when the two streams have the same mass flow (such as the case of make-up air systems), the expression for the effectiveness referred to as the efficiency) can be further simplified to:

$$\mathbf{\varepsilon = (T_{co} - T_{ci}) / (T_{hi} - T_{ci})}$$

Heat capacity ratio = $(T_{co} - T_{ci}) / (T_{hi} - T_{ho})$

If the effectiveness of a heat exchanger is 0.5, this does not mean that heat exchanger is only 50% efficient in its transfer of thermal energy. By conservation of energy, any energy that is lost on one side must be gained on the other so in that way we would say them as 100% efficient. But the effectiveness is actually just a measure of the ability of a heat exchanger to exchange temperatures.

If a perfect counter flow heat exchanger should be able to get the two fluids to swap temperatures (assuming the same fluid and mass flow rate). If a = 50 ^{0}C air and b = 90 ^{0}C air going through a perfect heat exchanger, then we should get a = 90 ^{0}C air and b = 50 ^{0}C air out of it. 50% effective would give 70 ^{0}C air out from both streams.

4.6 Examples

a) Liquid – Liquid Exchanger

(i) A shell and tube exchanger of following configuration is considered being used for oil cooler with oil at the shell side and cooling water at the tube side.

Tube Side

- 460 Nos x 25.4mmOD x 2.11mm thick x 7211mm long
- Pitch – 31.75mm 30^{o} triangular
- 2 Pass

Shell Side

- 787 mm ID
- Baffle space – 787 mm
- 1 Pass

The monitored parameters are as below:

Parameters	Units	Inlet	Outlet
Hot fluid flow, W	kg/h	719800	719800
Cold fluid flow, w	kg/h	881150	881150
Hot fluid Temp, T_h	^{O}C	145	102
Cold fluid Temp, T_c	^{O}C	25.5	49
Hot fluid Pressure, P	bar g	4.1	2.8
Cold fluid Pressure, p	bar g	6.2	5.1
Hot fluid specific heat Capacity, C_{ph}	kJ/(kg ^{O}C)	2.847	
Cold fluid specific heat Capacity, C_{pc}	kJ/(kg ^{O}C)	4.187	

Calculation of Thermal data:

Heat Transfer Area = 264.55 m^{2}

1. Heat Duty: $Q = q_s + q_l$

   Hot fluid, Q = 719800 x 2.847 x (145 –102) / 3600 = 24477.4 kW

   Cold Fluid, Q = 881150 x 4.187 x (49 – 25.5) / 3600 = 24083.4 kW

2. Hot Fluid Pressure Drop

   Pressure Drop = $P_i - P_o$ = 4.1 – 2.8 = 1.3 bar g.

3. Cold Fluid Pressure Drop

   Pressure Drop = $p_i - p_o$ = 6.2 – 5.1 = 1.1 bar g.

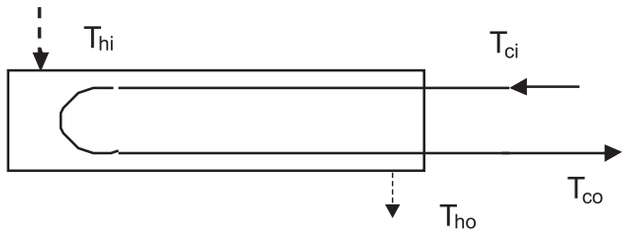
4. Temperature range hot fluid

   Temperature Range $\Delta T_h = T_{hi} - T_{ho}$ = 145 – 102 = 43 ^{o}C.

5. Temperature Range Cold Fluid

Temperature Range $\Delta T_c = T_{co} - T_{ci} = 49 - 25.5 = 23.5\ ^0C$.

6. LMTD



LMTD, Counter Flow = (96 – 76.5) / ln (96 / 76.5) = 85.9 ^{0}C.

7. LMTD Correction Factor, F, to account for Cross flow:

Computing the Parameters below:

R = (Ta-Tb) / (tb-ta) = (145-102) / (49-25.5) = 1.83

P = (tb-ta) / (Ta-ta) = (49-25.5) / (145-25.5) = 0.20

For R ≠ 1:

$\alpha = \{(1-RP) / (1-P)\}^{(1/N)} = \{(1- 1.83x0.2) / (1-0.2)\}^{(1/1)} = 0.793$

S = (0.793-1) / (0.793-1.83) = 0.20

$$F = \frac{\sqrt{R^2+1}\ \ln\left(\frac{1 - S}{1 - RS}\right)}{(R - 1)\ln\left(\frac{2 - S\,(R+1 - \sqrt{R^2+1}\,)}{2 - S\,(R+1 + \sqrt{R^2+1}\,)}\right)}$$

F = 0.977

8. Corrected LMTD = F x LMTD = 0.977 x 85.9 = 83.9 ^{o}C.

9. Overall Heat Transfer Co-efficient

U = Q/ (A x Corrected LMTD) = 24477.4/ (264.55 x 83.9) = 1.104 kW/m^2. K

Comparison of Calculated data with Design Data

Parameters	Units	Test Data	Design Data
Duty, Q	kW	24477.4	25623
Hot fluid side pressure drop, ΔP_h	Bar	1.3	1.34
Cold fluid side pressure drop, ΔP_c	Bar	1.1	0.95
Temperature Range hot fluid, ΔT_h	^oC	43	45
Temperature Range cold fluid, ΔT_c	^oC	23.5	25
Corrected LMTD, MTD	^oC	83.8	82.2
Heat Transfer Coefficient, U	**kW/(m^2. K)**	**1.104**	**1.178**

Heat Duty: Actual duty differences will be practically negligible as these duty differences could be because of the specific heat capacity deviation with the temperature. Also, there could be some heat loss due to radiation from the hot shell side.

Pressure drop: Also, the pressure drop in the shell side of the hot fluid is reported normal (only slightly less than the design figure). This is attributed with the increased average bulk temperature of the hot side due to decreased performance of the exchanger.

Temperature range: As seen from the data the deviation in the temperature ranges could be due to the increased fouling in the tubes (cold stream), since a higher pressure drop is noticed.

Heat Transfer coefficient: The estimated value has decreased due to increased fouling that has resulted in minimized active area of heat transfer.

Physical properties: If available from the data or Lab analysis can be used for verification with the design data sheet as a cross check towards design considerations.

(ii) In the above example 4.6(a), determine the Effectiveness of heat exchanger and Heat Capacity ratio.

Solution:

Hot fluid, Coil = (W x Cph) oil = (719800 x 2.847) / 3600 = 569.24 kW/ ^{o}C.

Cold fluid, Cwater = (w x Cpc) water = (881150 x 4.187) / 3600 = 1024.83 kW/ ^{o}C.

Therefore,

Cmin = 569.24

Cmax = 1024.83

Q = 24477.4 kW

ΔTmax = 145- 25.5 = 119.5 ^{o}C

Effectiveness of the heat exchanger, ε = Q / Qmax
= Q / (Cmin x ΔTmax)
= (24477.4) / (569.24 x 119.5)
= 0.3598 = 0.36

Heat Capacity Ratio, r = Cmin / Cmax
= 569.24 / 1024.83 = 0.55

b) A Plate Heat Exchanger with total heat transfer area of 41 m^2 is used to exchange heat between a hot effluent stream and cooling water stream.

The monitored parameters are given below:

Parameters	Unit	Inlet	Outlet
Hot fluid flow, W	kg/h	85200	85200
Hot fluid temperature, T_h	^{0} C	77	54
Cold fluid temperature, T_c	^{0} C	49	57

Calculate LMTD and Overall heat transfer coefficient, U, assuming LMTD correction factor of 0.9 for plate heat exchanger. Specific heat capacity of hot effluent stream is 4.187 kJ/kg^{0}C.

Solution:

Hot Load, Q = (85200 x 4.187 x (77-54)) / 3600 = 2279 kW.

LMTD, Counter flow = {(77-57) – (54-49)}/ {ln (77-57) / (54-49)}
= 10.8 ^{0} C

Correction Factor, F = 0.9 (given)

Corrected LMTD = F x LMTD
= 0.9 x 10.8 = 9.72 ^{0} C

Overall heat transfer coefficient, U = Q / (A x Corrected LMTD)
U = 2279 / (41 x 9.72) = 5.718 kW/m^2. ^{0} C

c) A double pipe heat exchanger is used to cool a hot stream from 177^0 C to 121^0 C by heating a cold stream from 49^0 C to 77^0 C. The hot stream will flow in the inner pipe in a counter flow arrangement to the cold stream in the outer pipe.

The heat transfer surface area of 18.5 m^2 will transfer the heat load of 1025.85 kW. Determine the overall heat transfer coefficient, U.

Solution:

LMTD, Counter flow = {(177-77) – (121-49)}/ {ln (177-77) / (121-49)}
= 85.2 ^{0} C

Overall heat transfer coefficient, U = Q / (A x LMTD)

= 1025.85 / (18.5 x 85.2)

= 0.651 kW/ m^2 ^{0} C

d) Surface Condenser

A shell and tube exchanger of following configuration is considered being used for Condensing turbine exhaust steam with cooling water at the tube side.

Tube Side
20648 Nos x 25.4mmOD x 1.22mm thk x 18300mm long

Pitch – 31.75mm 60^o triangular
1 Pass

The monitored parameters are as below:

Parameters	Units	Inlet	Outlet
Hot fluid flow, W	kg/h	939888	939888
Cold fluid flow, w	kg/h	55584000	55584000
Hot fluid Temp, T_h	^{O}C	No data	34.9
Cold fluid Temp, T_c	^{O}C	18	27
Hot fluid Pressure, P	Bar g	52.3 mbar	48.3
Cold fluid Pressure, p	Bar g	4	3.6

Calculation of Thermal data:

Area = 30151 m^2

1. Duty:

Hot fluid, Q = Mass flow rate of steam x Latent heat of vaporization

= (939888 / 3600) kg/sec x 2210 kJ/kg = 576990 kW

Cold Fluid, Q = mass flow rate x Cpc x temperature difference

= (55584000/3600) kg/sec x 4.187 kJ/kg ^{O}C x (27-18) ^{O}C

= 581825.5 kW

2. Hot Fluid Pressure Drop

Pressure Drop = $P_i - P_o$ = 52.3 – 48.3 = 4.0 mbar.

3. Cold Fluid Pressure Drop

Pressure Drop = $p_i - p_o$ = 4 – 3.6 = 0.4 bar.

4. Temperature range hot fluid

Temperature Range $\Delta T_h = T_{hi} - T_{ho}$ = No data

5. Temperature Range Cold Fluid

Temperature Range $\Delta T_c = T_{ci} - T_{co} = 27 - 18 = 9\ ^{o}C$.

6. LMTD

Calculated considering condensing part only (Hot fluid temperature at inlet and outlet remains same as 34.9 ^{O}C.)

LMTD, Counter Flow = ((34.9 – 18)-(34.9-27)) / ln ((34.9-18)/(34.9-27))
= 11.8 ^{0} C.

7. Correction Factor to account for Cross flow

F = 1.0

8. Corrected LMTD

MTD = F x LMTD = 1.0 x 11.8 = 11.8 ^{0} C.

9. Heat Transfer Co-efficient

Overall HTC, U = Q/ A ΔT = 576990/ (30151 x 11.8) = 1.622 kW/m^2.K

Comparison of Calculated data with Design Data

Parameters	Units	Test Data	Design Data
Duty, Q	kW	576990	588430
Hot fluid side pressure drop, ΔP_h	mBar	4 mbar	3.7 mbar
Cold fluid side pressure drop, ΔP_c	Bar	0.4	
Temperature Range hot fluid, ΔT_h	^{O}C		
Temperature Range cold fluid, ΔT_c	^{O}C	(27-18) = 9	(28-19)=9
Corrected LMTD, MTD	^{O}C	11.8	8.9
Heat Transfer Coefficient, U	kW/(m^2. K)	1.622	2.37

Heat Duty: Actual duty differences will be practically negligible as these duty differences could be because of the specific heat capacity deviation with the temperature. Also, there could be some heat loss due to radiation from the hot shell side.

Pressure drop: The condensing side operating pressure raised due to the backpressure caused by the non-condensable. This has resulted in increased pressure drop across the steam side.

Temperature range: With reference to cooling waterside there is no difference in the range however, the terminal temperature differences has increased indicating lack of proper heat transfer.

Heat Transfer coefficient: Heat transfer coefficient has decreased due to increased amount of non-condensable with the steam.

Trouble shooting:

Operations may be checked for tightness of the circuit and ensure proper venting of the system. The vacuum source might be verified for proper functioning.

Solved Example:

Flow rates of the hot and the cold water streams flowing through a heat exchanger are 12 and 30 kg/min, respectively. Hot and cold water stream inlet temperatures are 72 °C and 27 °C, respectively. The exit temperature of the hot stream is required to be 52 °C. The specific heat of water is 4.179 kJ/kg K. The overall heat transfer coefficient is 800 W/m^2 K.

Neglecting the effect of fouling, calculate the heat transfer area for

a) Parallel-flow
b) Counter-flow

Solution:

a)

Rate of heat transfer	M x cp x delt T (12/60) × 4.179 x 1000 × (72 – 52) = 16716 W
Cold water exit temperature	27 + [16716 / (4.179 x 1000 × (30/60))]
	27 + 8 = 35 °C

Terminal temperature differences for parallel flow heat exchangers	(72 – 27) and (52 – 35) ^{0}C i.e., 45 °C and 17 °C respectively.
LMTD	(45 – 17)/ln(45/17) = 28.76
Overall heat transfer coefficient U	800 W/m^2 K
Heat transfer area required for parallel flow	[16716 / (800 × 28.76)]
	= 0.72 m^2

b)

Terminal temperature differences for counter flow heat exchangers	(72 – 35) and (52 – 27) ^{0}C i.e., 37 °C and 25 °C respectively.
LMTD	(37 – 25)/ln(37/25) = 30.6
Overall heat transfer coefficient U	800 W/m^2 K
Heat transfer area required for counter flow	[16716 / (800 × 30.6)]
	0.684 m^2

QUESTIONS	
S-1	Write the overall heat transfer coefficient U, as a function of sensible heat (qs) and latent heat (qL).
S-2	In a shell and tube heat exchanger, engaged in heat transfer between fouling fluid and clear fluid, the fouling fluid should be put on shell side or tube side?
S-3	Explain the terms heat duty and capacity ratio.
S-4	What is meant by fouling?
S-5	Explain why the term effectiveness rather than efficiency is used in the performance assessment of heat exchanger.
S-6	Explain with sketch how to determine the LMTD of a Counter flow heat exchanger
S-7	In a heat exchanger the hot stream enters at 70^{O}C and leaves at 55^{O}C. On the other side the cold stream enters at 30^{O}C and leaves at 55^{O}C. Find out the LMTD of the heat exchanger if it is counter flow type.
S-8	In a multi pass shell and tube heat exchanger what is the need for correction factor while estimating LMTD.
S-9	What is the significance of overall heat transfer coefficient?
S-10	Write the unit of overall heat transfer coefficient.
L-1	The flow rates of the hot and the cold water streams flowing through a heat exchanger are 10 and 25 kg/min, respectively. Hot and cold side inlet temperatures are 70°C and 25°C, respectively. The other data is given below. (i) effect of fouling can be neglected, (ii) the exit temperature of the hot side stream is required to be 50°C, and (iii) the overall heat transfer coefficient is 800 W/m^2 K (iv) specific heat of water is 4.179 kJ/kg K. a) Calculate the heat transfer area of the heat exchanger if the heat exchanger is parallel-flow b) Calculate the heat transfer area of the heat exchanger if the heat exchanger is counter-flow
L-2	In a heat exchanger, the hot stream enters at 80°C and leaves at 50°C. On the other hand, the cold streams enters at 20°C and leaves the heat exchanger at 50°C. Determine whether the heat exchanger is counter-current type or co-current type.

N-1	In an air cooled heat exchanger hot fluid (specific heat: 1 kCal/kg ^{o}C) is entering at a temperature of 80 ^{0}C and leaving at a temperature of 38 ^{0}C. Flow rate of the hot fluid is 63450 kg/hr. Air is entering at a temperature of 30 ^{0}C and leaving at a temperature of 60 ^{0}C. Flow rate of the air is 370057 kg/hr. Power drawn by the fan is 30 kW. The plant persons want to replace it with a water cooled counter flow plate heat exchanger. Given data: Annual operating hours : 4800 hrs Pump Efficiency : 75% Motor efficiency : 90 % Effectiveness of water cooled heat exchanger is 0.4 Water is available at 25 ^{O}C Total head developed by the pump is 4 kg/Cm^2
	Over all heat transfer coefficient of PHE is 22300 kCal/hr/m^2/^{0}C For water cooled system the additional fan power consumption is 5 kW. Calculate: a) Saving due to replacement b) Area of the plate heat exchanger

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Chapter 5

Performance Evaluation of Electric motors and Variable Speed Drives

Learning Objectives

In this chapter you will learn about

- Methods for determining motor loading
- Method of determining motor efficiency
- Evaluating performance of rewound motors
- Variable speed drive: principles and applications
- Factors for successful implementation of variable speed drive

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Evaluate the efficiency of induction motor in the field**
- ✓ **Conduct a motor loading survey**
- ✓ **Identify motors for reshuffling/ replacement**
- ✓ **Check performance of rewound motors**
- ✓ **Investigate application potential for variable speed drive**

5. ENERGY PERFORMANCE ASSESSMENT OF MOTORS AND VARIABLE SPEED DRIVES

5.1 Introduction

The two parameters of importance in a motor are efficiency and power factor. The efficiencies of induction motors remain almost constant between 50% to 100% loading (Refer figure 5.1). With motors designed to perform this function efficiently; the opportunity for savings with motors rests primarily in their selection and use. When a motor has a higher rating than that required by the equipment, motor operates at part load. In this state, the efficiency of the motor is reduced. Replacement of under loaded motors with smaller motors will allow a fully loaded smaller motor to operate at a higher efficiency. This arrangement is generally most economical for larger motors, and only when they are operating at less than one-third to one-half capacity, depending on their size.

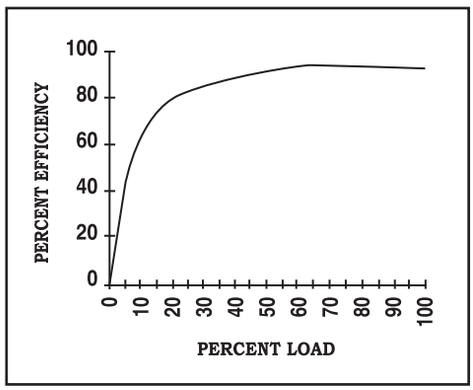


Figure 5.1 Efficiency vs. Loading

5.2 Performance Terms and Definitions

Efficiency :

The efficiency of the motor is given by

$$\eta = \frac{P_{out}}{P_{in}} = 1 - \frac{P_{loss}}{P_{in}}$$

Where P_{out} – Output power of the motor
P_{in} – Input power of the motor
P_{Loss} – Losses occurring in motor

Motor Loading :

$$Motor\ Loading\ \% = \frac{Actual\ operating\ load\ of\ the\ motor}{Rated\ capacity\ of\ the\ motor}\ x\ 100$$

5.3 Efficiency Testing

While input power measurements are fairly simple, measurement of output or losses need a laborious exercise with extensive testing facilities. The following are the testing standards widely used.

Europe: IEC 60034-2, and the new IEC 61972
US: IEEE 112 - Method B
Japan: JEC 37

Even between these standards the difference in efficiency value is up to 3%.

For simplicity nameplate efficiency rating may be used for calculations if the motor load is in the range of 50 –100 %.

Field Tests for Determining Efficiency

(Note: The following section is a repeat of material provided in the chapter-2 on Electrical Motors in Book-3.)

No Load Test :

The motor is run at rated voltage and frequency without any shaft load. Input power, current, frequency and voltage are noted. The no load P.F. is quite low and hence low PF watt meters are required. From the input power, stator I^2R losses under no load are subtracted to give the sum of Friction and Windage (F&W) and core losses. To separate core and F & W losses, test is repeated at variable voltages. It is worthwhile plotting no-load input kW versus Voltage; the intercept is F & W kW loss component.

F&W and core losses = No load power (Watts) – (No load current)^2 x Stator resistance

Stator and Rotor I^2R Losses :

The stator winding resistance is directly measured by a bridge or volt amp method. The resistance must be corrected to the operating temperature. For modern motors, the operating temperature is likely to be in the range of 100^{0}C to 120^{0}C and necessary correction should be made. Correction to 75^{0}C may be inaccurate. The correction factor is given as follows :

$$\frac{R_2}{R_1} = \frac{235 + t_2}{235 + t_1}$$, where, t_1 = ambient temperature, ^{o}C & t_2 =- operating temperature, ^{o}C.

The rotor resistance can be determined from locked rotor test at reduced frequency and reduced voltage, but rotor I^2R losses are measured from measurement of rotor slip.

Rotor I^2R losses = Slip × (Stator Input – Stator I^2R Losses – Core Loss)

Accurate measurement of slip is possible by stroboscope or non-contact type tachometer. Slip also must be corrected to operating temperature.

Stray Load Losses :

These losses are difficult to measure with any accuracy. IEEE Standard 112 gives a complicated method, which is rarely used on shop floor. IS and IEC standards take a fixed value as 0.5 % of output. It must be remarked that actual value of stray losses is likely to be more. IEEE – 112 specifies values from 0.9 % to 1.8 %.

Motor Rating	Stray Losses
1 – 125 HP	1.8 %
125 – 500 HP	1.5 %
501 – 2499 HP	1.2 %
2500 and above	0.9 %

Points for Users :

It must be clear that accurate determination of efficiency is very difficult. The same motor tested by different methods and by same methods by different manufacturers can give a difference of 2 %.

Estimation of efficiency in the field can be summarized as follows:

a) Measure stator resistance and correct to operating temperature. From rated current value, I^2R losses are calculated.

b) From rated speed and output, rotor I^2R losses are calculated

c) From no load test, core and F & W losses are determined for stray loss

The method is illustrated by the following example :

Example :

Motor Specifications

Rated power	=	34 kW/45 HP
Voltage	=	415 Volt
Current	=	57 Amps
Speed	=	1475 rpm
Insulation class	=	F
Frame	=	LD 200 L
Connection	=	Delta

No load test Data

Voltage, V	=	415 Volts
Current, I	=	16.1 Amps
Frequency, F	=	50 Hz
Stator phase resistance at 30^0C	=	0.264 Ohms
No load power, P_{nl}	=	1063.74 Watts

a) Calculate iron plus friction and windage losses

b) Calculate stator resistance at 120^0C

$$R_2 = R_1 \times \frac{235 + t_2}{235 + t_1}$$

c) Calculate stator copper losses at operating temperature of resistance at 120^0C

d) Calculate full load slip(s) and rotor input assuming rotor losses are slip times rotor input.

e) Determine the motor input assuming that stray losses are 0.5 % of the motor rated power

f) Calculate motor full load efficiency and full load power factor

Solution

a) Let Iron plus friction and windage loss, P_i + fw

No load power, Pnl = 1063.74 Watts

Stator Copper loss, P st-30^0C (Pst.cu)
= $3 \times (16.1 / \sqrt{3})^2 \times 0.264$
= 68.43 Watts

Pi + fw = P_{nl} – Pst.cu
= 1063.74 – 68.43
= 995.3 W

b) Stator Resistance at 120^0C,

$$R_{120\ ^0C} = 0.264 \times \frac{120 + 235}{30 + 235}$$

= 0.354 ohms per phase

c) Stator copper losses at full load, Pst.cu 120^0C

$= 3 \times (57 / \sqrt{3})^2 \times 0.354$

= 1150.1 Watts

d) Full load slip

S = (1500 – 1475) / 1500

= 0.0167

Rotor input, Pr = P_{output}/ (1-S)

= 34000 / (1-0.0167)

= 34577.4 Watts

e) Motor full load input power, P input

$= P_r$ + Pst.cu 120^0C + (P_i + fw) + P_{stray}

= 34577.4 + 1150.1 + 995.3 + ($0.005^* \times 34000$)

= 36892.8 Watts

*where, stray losses = 0.5% of rated output (assumed)

f) Motor efficiency at full load

$$\text{Efficiency} = \frac{P_{output}}{P_{input}} \times 100$$

$$= \frac{34000}{36892.8} \times 100$$

= 92.2 %

$$\text{Full Load PF} = \frac{P_{input}}{\sqrt{3} \times V \times I_f}$$

$$= \frac{36892.8}{\sqrt{3} \times 415 \times 57}$$

= 0.90

Comments :

a) The measurement of stray load losses is very difficult and not practical even on test beds.

b) The actual value of stray loss of motors up to 200 HP is likely to be 1 % to 3 % compared to 0.5 % assumed by standards.

c) The value of full load slip taken from the nameplate data is not accurate. Actual measurement under full load conditions will give better results.

d) The friction and windage losses really are part of the shaft output; however, in the above calculation, it is not added to the rated shaft output, before calculating the rotor input power. The error however is minor.

e) When a motor is rewound, there is a fair chance that the resistance per phase would increase due to winding material quality and the losses would be higher. It would be interesting to assess the effect of a nominal 10 % increase in resistance per phase.

5.4 Determining Motor Loading

1. By Input Power Measurements

- First measure input power $\mathbf{P}_i$ with a hand held or in-line power meter
- P_i = Three-phase power in kW
- Note the rated kW and efficiency from the motor name plate
- The figures of kW mentioned in the name plate is for output conditions. So corresponding input power at full-rated load

$$P_{ir} = \frac{\text{Nameplate full rated kW}}{\eta_{fl}}$$

η_{fl} = Efficiency at full-rated load

P_{ir} = Input power at full-rated load in kW

The percentage loading can now be calculated as follows

$$Load = \frac{P_i}{P_{ir}} \; x \; 100\%$$

Example

The nameplate details of a motor are given as power = 15 kW, efficiency η = 0.9. Using a power meter the actual three phase power drawn is found to be 8 kW. Find out the loading of the motor.

Input power at full-rated power in kW, P_{ir} = 15 /0.9
= 16.7 kW

Percentage loading = 8/16.7
= 48 %

2. By Line Current Measurements

The line current load estimation method is used when input power cannot be measured and only amperage measurements are possible. The amperage draw of a motor varies approximately linearly with respect to load, down to about 75% of full load. Below the 75% load point, power factor degrades and the amperage curve becomes increasingly non-linear. **In the low load region, current measurements are not a useful indicator of load.** However, this method may be used only as a preliminary method just for the purpose of identification of oversized motors.

$$\%Load = \frac{Input\ load\ current}{Input\ rated\ current} * 100 \text{ (Valid up to 75\% loading)}$$

3. Slip Method

In the absence of a power meter, the slip method can be used which requires a tachometer. This method also does not give the exact loading on the motors.

$$Load = \frac{Slip}{S_s - S_r} * 100\%$$

Where,

Load = Output power as a % of rated power
Slip = Synchronous speed - Measured speed in rpm
S_s = Synchronous speed in rpm at the operating frequency
S_r = Nameplate full-load speed

Example: Slip Load Calculation

Given: Synchronous speed in rpm = 1500 at 50 HZ operating frequency.
(Synchronous speed = 120f/P) ; f: frequency, P: Number of poles
Nameplate full load speed = 1450
Measured speed in rpm = 1480
Nameplate rated power = 7.5 kW

Determine actual output power. $Load = \frac{1500-1480}{1500-1450} * 100\% = 40\%$

From the above equation, actual output power would be 40% *x 7.5* kW = 3 kW

The speed/slip method of determining motor part-load is often favored due to its simplicity and safety advantages. Most motors are constructed such that the shaft is accessible to a tachometer or a strobe light.

The accuracy of the slip method, however, is limited. The largest uncertainty relates to the accuracy with which manufacturers report the nameplate full-load speed. Manufacturers generally round their reported full-load speed values to some multiple of 5 rpm. While 5 rpm is but a small percent of the full-load speed and may be considered as insignificant, the slip method relies on the difference between full-load nameplate and synchronous speeds. Given a 40 rpm "correct" slip, a seemingly minor 5 rpm disparity causes a 12% change in calculated load.

Slip also varies inversely with respect to the motor terminal voltage squared. A voltage correction factor can, also, be inserted into the slip load equation. The voltage compensated load can be calculated as shown

$$Load = \frac{Slip}{(S_s - S_r) \, x \, (V_r / V)^2} \, x \, 100\%$$

Where,

Load = Output power as a % of rated power

Slip = Synchronous speed - Measured speed in rpm

S_s = Synchronous speed in rpm

S_r = Nameplate full-load speed

V = RMS voltage, mean line to line of 3 phases

V_r = Nameplate rated voltage

5.5 Performance Evaluation of Rewound Motors

Ideally, a comparison should be made of the efficiency before and after a rewinding. A relatively simple procedure for evaluating rewind quality is to keep a log of no-load input current for each motor in the population. This figure increases with poor quality rewinds. A review of the rewind shop's procedure should also provide some indication of the quality of work. When rewinding a motor, if smaller diameter wire is used, the resistance and the I^2R losses will increase.

5.6 Format for Data Collection

The motor loading survey can be performed using the format given below:

Motor Field Measurement Format

Company_____ Location_____

Date _____ Process_____

Department_____

General Data

Driven Equipment_____

Motor Name Plate Data

Manufacturer _____

Model _____

Serial Number _____

Type :Squirrel cage/Slip ring_____

Size (hp/kW)_____

Synchronous Speed (RPM) _____

Full-Load Speed (RPM) _____

Voltage Rating _____

Full-Load Amperage _____

Full-Load Power Factor (%) _____

Full-Load Efficiency (%) _____

Temperature Rise _____

Insulation Class _____

From Test Certificate

Load	100%	75%	25%	No Load
Current				
PF				
Efficiency				

Stator resistance per phase =

Rewound ☐ Yes ,if yes How many times rewound ?---

☐ No

Motor Loading % _____

Motor Operating Profile:

No of hours of operation

I Shift _____

II Shift _____

III Shift _____

Annual Operating Time _____ hours/year

Type of load

1.Load is quite steady, motor "On" during shift ☐
2.Load starts, stops, but is constant when "On" ☐
3.Load starts, stops, and fluctuates when "On" ☐

Measured Data

Supply Voltage
By Voltmeter
V_{RY} _____
V_{YB} _____ V avg _____
V_{BR} _____

Input Amps
By Ammeter
A a _____
A b _____ A avg _____
A c _____

Power Factor (PF) _____
Input Power (kW) _____

Motor Operating Speed _____RPM
At frequency of _____

Driven Equipment Operating Speed

_____RPM

Type of Transmission (Direct/Gear/Fluid coupling)

The monitoring format for rewound motor is given below:

Section	Equipment Code	Motor Code	Motor Type		No Load Current				Starter Resistance/phase		No load loss	
			Sq.Cage	Slip Ring	New Motor		After Rewinding		New	Rewound	New	Rewound
					A	V	A	V			Watts	Watts

5.7 Application of Variable Speed Drives (VSD)

Although there are many methods of varying the speeds of the driven equipment such as hydraulic coupling, gear box, variable pulley etc., the most possible method is one of varying the motor speed itself by varying the frequency and voltage by a variable frequency drive.

5.7.1 Concept of Variable Frequency Drive

The speed of an induction motor is proportional to the frequency of the AC voltage applied to it, as well as the number of poles in the motor stator. This is expressed by the equation:

$$RPM = (f \times 120) / p$$

Where f is the frequency in Hz, and p is the number of poles in any multiple of 2.

Therefore, if the frequency applied to the motor is changed, the motor speed changes in direct proportion to the frequency change. The control of frequency applied to the motor is the job given to the VSD.

The VSD's basic principle of operation is to convert the electrical system frequency and voltage to the frequency and voltage required to drive a motor at a speed other than its rated speed. The two most basic functions of a VSD are to provide power conversion from one frequency to another, and to enable control of the output frequency.

VSD Power Conversion

As illustrated by Figure 5.2, there are two basic components, a rectifier and an inverter, to accomplish power conversion.

The rectifier receives the 50-Hz AC voltage and converts it to direct current (DC) voltage. A DC bus

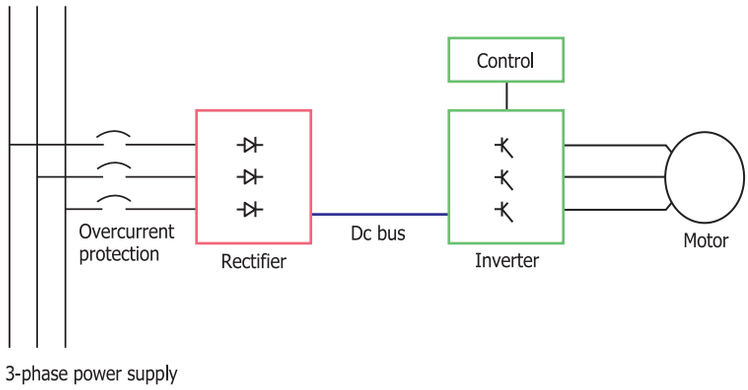


Figure 5.2 Components of a Variable Speed Drive

inside the VSD functions as a "parking lot" for the DC voltage. The DC bus energizes the inverter, which converts it back to AC voltage again. The inverter can be controlled to produce an output frequency of the proper value for the desired motor shaft speed.

5.7.2 Factors for Successful Implementation of Variable Speed Drives

a) Load Type for Variable Frequency Drives

The main consideration is whether the variable frequency drive application requires a **variable torque or constant torque** drive. If the equipment being driven is centrifugal, such as a fan or pump, then a variable torque drive will be more appropriate. Energy savings are usually the primary motivation for installing variable torque drives for centrifugal applications. For example, a fan needs less torque when running at 50% speed than it does when running at full speed. Variable torque operation allows the motor to apply only the torque needed, which results in reduced energy consumption.

Conveyors, positive displacement pumps, punch presses, extruders, and other similar type applications require constant level of torque at all speeds. In which case, constant torque variable frequency drives would be more appropriate for the job. A constant torque drive should have an **overload current capacity** of 150% or more for one minute. Variable torque variable frequency drives need only an overload current capacity of 120% for one minute since centrifugal applications rarely exceed the rated current.

If tight process control is needed, then you may need to utilize a sensor less vector, or flux vector variable frequency drive, which allow a high level of accuracy in controlling speed, torque, and positioning.

b) Motor Information

The following motor information will be needed to select the proper variable frequency drive:

Full Load Amperage Rating. Using a motor's horsepower is an inaccurate way to size variable frequency drives.

Speed Range. Generally, a motor should not be run at any speed less than 20% of its specified maximum speed allowed. If it is run at a speed less than this without auxiliary motor cooling, the motor will overheat. Auxiliary motor cooling should be used if the motor must be operated at very slow speeds.

Multiple Motors. To size a variable frequency drive that will control more than one motor, add together the full-load amp ratings of each of the motors. All motors controlled by a single drive must have an equal voltage rating.

c) Efficiency and Power Factor

The variable frequency drive should have an efficiency rating of **95% or better at full load**.

Variable frequency drives should also offer a **true system power factor of 0.95 or better** across the operational speed range, to save on demand charges, and to protect the equipment (especially motors).

d) Protection and Power Quality

Motor overload Protection for instantaneous trip and motor over current.

Additional Protection: Over and under voltage, over temperature, ground fault, control or microprocessor fault. These protective circuits should provide an orderly shutdown of the VFD, provide indication of the fault condition, and require a manual reset (except under voltage) before restart. Under voltage from a power loss shall be set to automatically restart after return to normal. The history of the previous three faults shall remain in memory for future review.

If a built-up system is required, there should also be externally-operated short circuit protection, door-interlocked fused disconnect and circuit breaker or motor circuit protector (MCP)

To determine if the equipment under consideration is the right choice for a variable speed drive:

The load patterns should be thoroughly studied before exercising the option of VSD. In effect the load should be of a varying nature to demand a VSD (refer figure 5.3 & 5.4).

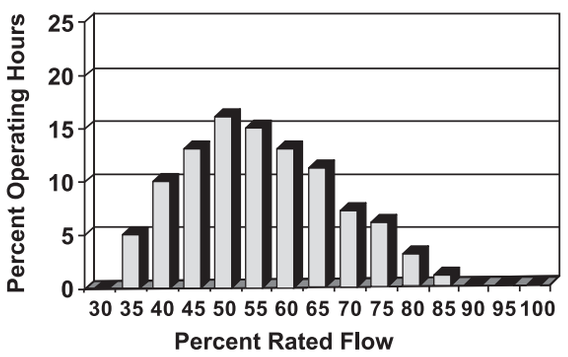


Figure 5.3 Example of an excellent variable speed drive candidate

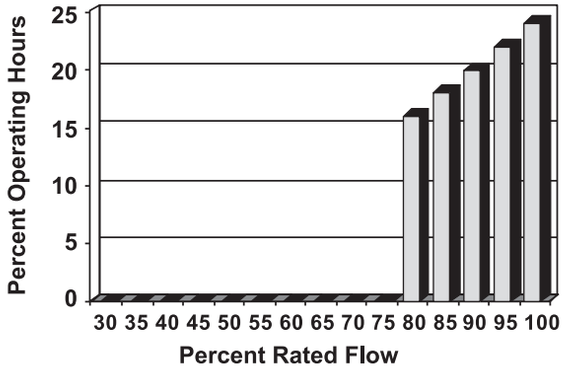


Figure 5.4 Example of a poor variable speed drive candidate

The first step is to identify the number of operating hours of the equipment at various load conditions. This can be done by using a Power analyzer with continuous data storage or by a simple energy meter with periodic reading being taken.

5.7.3 Information needed to Evaluate Energy Savings for Variable Speed Application

1. Method of flow control to which adjustable speed is compared:
   - output throttling (pump) or dampers (fan)
   - recirculation (pump) or unrestrained flow (fan)
   - adjustable-speed coupling (eddy current coupling)
   - inlet guide vanes or inlet dampers (fan only)
   - two-speed motor.

2. Pump or fan data:
   - head v's flow curve for every different type of liquid (pump) or gas (fan) that is handled
   - Pump efficiency curves.

3. Process information:
   - specific gravity (for pumps) or specific density of products (for fans)
   - system resistance head/flow curve
   - equipment duty cycle, i.e. flow levels and time duration.

4. Efficiency information on all relevant electrical system apparatus:
   - motors, constant and variable speed
   - variable speed drives
   - gears
   - transformers.

If we do not have precise information for all of the above, we can make reasonable assumptions for points 2 and 4.

Solved Example:

a) In a 75 kW four pole induction motor operating at 49.8 Hz and rated for 415 V and 1440 RPM, the actual measured speed is 1470 RPM. Find out the percentage loading of the motor if the voltage applied is 428 V.

b) A 6 pole, 415 volt, 3 Φ, 50 Hz induction motor delivers 22 kW power at rotor shaft at a speed of 950 rpm with PF of 0.88. The total loss in the stator including core, copper and other losses is 2 kW. Calculate the following.

   i) Slip

   ii) Rotor Copper Loss

   iii) Total Input to motor

   iv) Line current at 415 V and motor pf of 0.88

   v) Motor operating efficiency

Solution:

a)

$$\% \text{ Loading} = \frac{\text{Slip}}{(Ss - Sr) \times (Vr / V)^2} \times 100\%$$

Synchronous speed = 120 x 49.8 / 4 = 1494 rpm

Slip = Synchronous Speed – Measured speed in rpm.
= 1494 – 1470 = 24 rpm.

$$\% \text{ Loading} = \frac{24}{(1494 - 1440) \times (415/428)^2} \times 100\% = 47.27\%$$

b)

Synchronous Speed = (120 x 50 / 6) = 1000 rpm

Motor Speed = 950 rpm

(i) Slip = (1000 – 950) / 1000 = **5 %**

Power input to rotor = {(22 / (1 – 0.05)} = 23.16 kW

(ii) Rotor Copper Loss = (0.05 x 23.16) = 1.158 kW

Or = 23.16 - 2 2 = **1.16 kW**

(iii) Total Input to motor = (23.16 + 2) = **25.16 kW**

(iv) Line Current = (25.16 x 1000) /(√3 x 415 x0.88)

= **39.75 Amps**

Motor Efficiency = (22 / 25.16) = **87.44 %**

QUESTIONS	
S-1	What parameters are required to be measured to determine motor efficiency?
S-2	The resistance of a motor stator winding at 30°C is 0.264 ohms per phase. What will be the resistance of the stator winding per phase when the motor winding temperature is 100°C?
S-3	Calculate the percentage loading of a motor using the following data: Synchronous speed in rpm = 1500 at 50 HZ operating frequency. Nameplate full load speed = 1440 Measured speed in rpm = 1475 Nameplate rated power = 11 kW
S-4	Calculate the rotor I^2 R losses using the following data: Slip = 4% Stator input = 3000 W Stator I^2R losses = 100 W Core loss = 50 W
S-5	What are the two main advantages of application of a variable frequency drive in an induction motor other than accurate speed regulation?
S-6	What are the two factors influencing the speed of induction motor?
S-7	A 4 pole motor is operating at a frequency of 50 Hz. What is the synchronous speed of the motor?
S-8	If no instrument other than tachometer is available, what method you would suggest for measuring the motor load?
S-9	Which loss is assumed in the efficiency determination of an induction motor?
S-10	Why the line current method used for estimating loading of a motor is not applicable for motor loading less than 75%.
L-1	On the recommendations of energy auditor a company replaced a 15 kW, 4 pole standard motor by a 15 kW, 4 pole energy efficient motor for a centrifugal fan. The power consumption of energy efficient motor actually increased, while the grid frequency and voltage remained same. What could be the reason?
L-2	Determine the actual output power of an induction motor using the following data? No. of poles = 2 Frequency = 50 Hz Rated voltage = 415 V Name plate full speed = 2980 rpm Measured speed at 423 V = 2990 rpm Name plate rated power = 22 kW
N-1	Describe the various methods by which you calculate motor loading.

REFERENCES

1. Motor challenge: Office of Industrial Technologies, Department of Energy, USA

2. Energy audit Reports of National Productivity Council

Chapter 6

Performance Evaluation of Fans and Blowers

Learning Objectives

In this chapter you will learn about

- Performance terms and definitions
- Flow and pressure measurement location
- Determination of traverse points for velocity measurements in ducts
- Instruments for velocity measurement: Pitot tube and anemometer
- Static pressure, dynamic pressure and total pressure
- Density correction, velocity and flow measurement
- Method of assessing the fan efficiency

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Measure flows and pressures in closed ducts**
- ✓ **Correct density as per site condition**
- ✓ **Evaluate efficiency of fan**

6. ENERGY PERFORMANCE ASSESSMENT OF FANS AND BLOWERS

6.1 Introduction

This section describes the method of testing a fan installed on site in order to determine the performance of the fan in conjunction with the system to which it is connected.

6.2 Purpose of the Performance Test

The purposes of such a test are to determine, under actual operating conditions, the volume flow rate, the power input and the static pressure rise across the fan.

These test results will provide actual flow resistance of the air duct system, which can be compared with the design value of fan specified by supplier.

6.3 Performance Terms and Definitions

Static Pressure: The absolute pressure at a point minus the reference atmospheric pressure.

Dynamic Pressure: The rise in static pressure which occurs when air moving with specified velocity at a point is bought to rest without loss of mechanical energy. It is also known as velocity pressure.

Total Pressure: The sum of static pressures and dynamic pressures at a point.

Fan Shaft Power: The mechanical power supplied to the fan shaft

Motor Input Power: The electrical power supplied to the terminals of an electric motor drive.

Fan Efficiency = Air power / Fan shaft power

Static Fan Efficiency, % $= \dfrac{\text{Volume in } m^3/\text{sec} \times \Delta p \text{ Static pressure, mmWC)}}{102 \times \text{Fan shaft power (kW)}} \times 100$

Δp Static pressure = Discharge static pressure, mm WC – Suction static pressure, mmWC

6.4 Scope

The procedure describes field testing of centrifugal fans and blowers for assessing performance and efficiency.

6.5 Reference Standards

British Standard, BS 848 - Fans for general purposes Part 1, Methods of testing performance.

6.6 Field Testing

6.6.1 Instruction for Site Testing

Before site tests are carried out, it should be ensured that:

- Fan and its associated equipment are functioning properly, and at the rated speed
- Operations are at stable conditions, i.e. steady temperatures, densities, system resistance etc.

6.6.2 Location of Measurement Planes

General: The flow measurement plane shall be located in any suitable straight length, (preferably on the inlet side of the fan) where the airflow conditions are substantially axial, symmetrical and free from turbulence. Leakage of air from or into the air duct shall be negligible between the flow measuring plane and the fan. Bends and obstructions in an air duct can disturb the airflow for a considerable distance downstream, and should be avoided for the purposes of the test.

Test length: That part of the duct in which the flow measurement plane is located, is termed the 'test length' and shall be straight, of uniform cross section and free from any obstructions which may modify the airflow. It shall have a length equal to not less than twice the equivalent diameter of the air duct (i.e. $2D_e$). For rectangular duct, equivalent diameter, D_e is given by 2 LW/ (L+W) where L, W is the length and width of the duct. For circular ducts D_e is the same as diameter of the duct.

Inlet side of the fan: Where the 'test length' is on the inlet side of the fan, its downstream end shall be at a distance from the fan inlet equal to atleast $0.75D_e$. See figure 6.1. In the case of a fan having an inlet box , the downstream end of the test length shall be at a distance from the nearest part of the inlet cone of the fan equal to at least $0.75D_e$.

Outlet side of the fan: Where the 'test length' is on the outlet side of the fan, the upstream end of the 'test length' shall be at a distance from the fan outlet of at least $3D_e$. See figure 6.2. For this purpose, the fan outlet shall be considered as being the outlet of any expander on the outlet side of the fan.

Location of the Flow Measurement Plane within 'Test Length' : The flow measurement plane shall be located within the 'test length' at a distance from the downstream end of the 'test length' equal to at least 1.25 De.

Location of Pressure Measurement Plane : For the purpose of determining the pressure rise produced by the fan, the static pressure shall be measured at planes on the inlet and/or the outlet side of the fan sufficiently close to it to ensure that the pressure losses between the measuring planes and the fan are calculable in accordance with available friction factor data without adding excessively to the uncertainty of fan pressure determination.

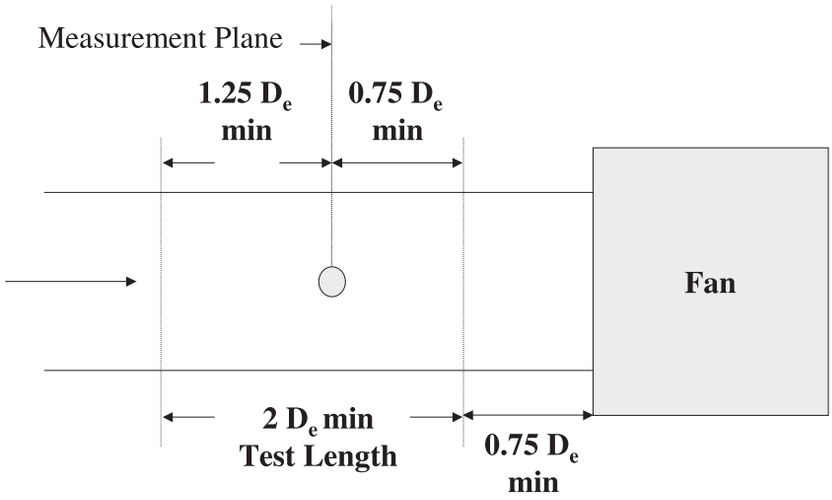


Figure 6.1 Test length on inlet side of fan

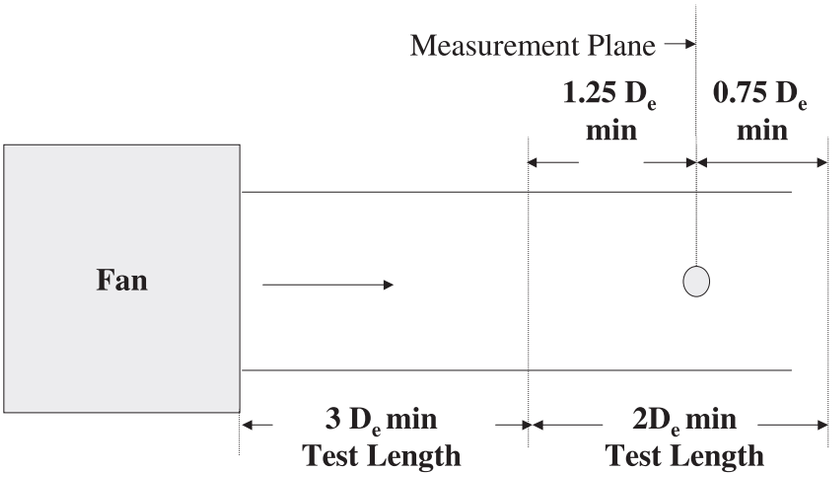


Figure 6.2 Test length on outlet side of fan

If conveniently close to the fan, the 'test length' selected for air flow measurement should also be used to pressure measurement. Other planes used for pressure measurement should be no closer than $0.25D_e$ from the fan inlet and no closer than $4D_e$ from the fan outlet. The plane of pressure measurement should be selected at least $4D_e$ downstream of any bend, expander or obstruction which are likely to cause separated flow or otherwise interfere with uniformity of pressure distribution.

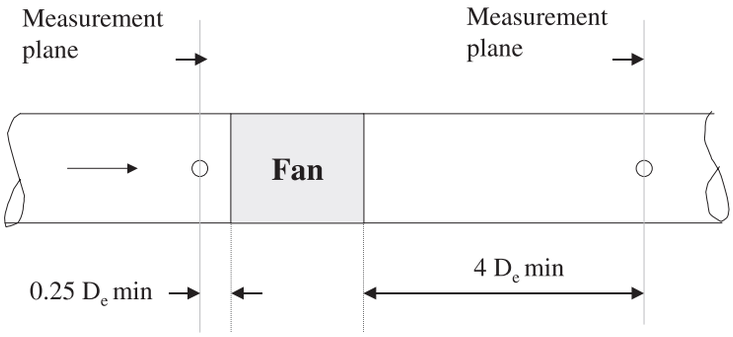


Figure 6.3 Location of pressure measurement planes for site testing

6.6.3 Measurement of Air Velocity on Site

Velocity shall be measured by either pitot tube or a rotating vane anemometer. When in use, the tubes connecting the pitot tube to pressure measuring instrument should be air tight. The calibration of pressure measuring instrument should be up to date.

Pitot Tube: In Figure 6.4, note that separate static connections (A) and total pressure connections (B) can be connected simultaneously across a manometer (C). Since the static pressure is applied to both sides of the manometer, its effect is cancelled out and the manometer indicates only the velocity pressure.

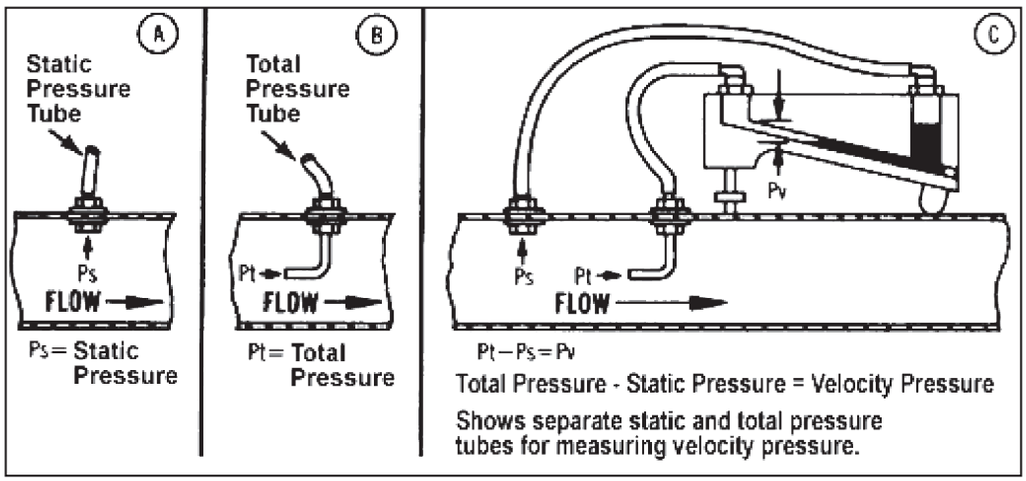


Figure 6.4 Types of Pressure Measurement

Static pressure and velocity pressure measurements are shown in Figure 6.5. To ensure accurate velocity pressure readings, the Pitot tube nozzle tip must be pointed directly into and parallel with the air stream.

In practice this type of measurement is usually made with a Pitot tube which incorporates both static and total pressure sensors in a single unit. Essentially, a Pitot tube consists of an impact tube (which receives total pressure input) fastened concentrically inside a second tube of slightly larger diameter which receives static pressure input from radial sensing holes around the tip. The air space between inner and outer tubes permits transfer of pressure from the sensing holes to the static pressure connection at the opposite end of the Pitot and then, through connecting tubing, to the low or negative pressure side of a manometer. When the total pressure tube is connected to the high pressure side of the manometer, velocity pressure is indicated directly. See Figure 6.5.

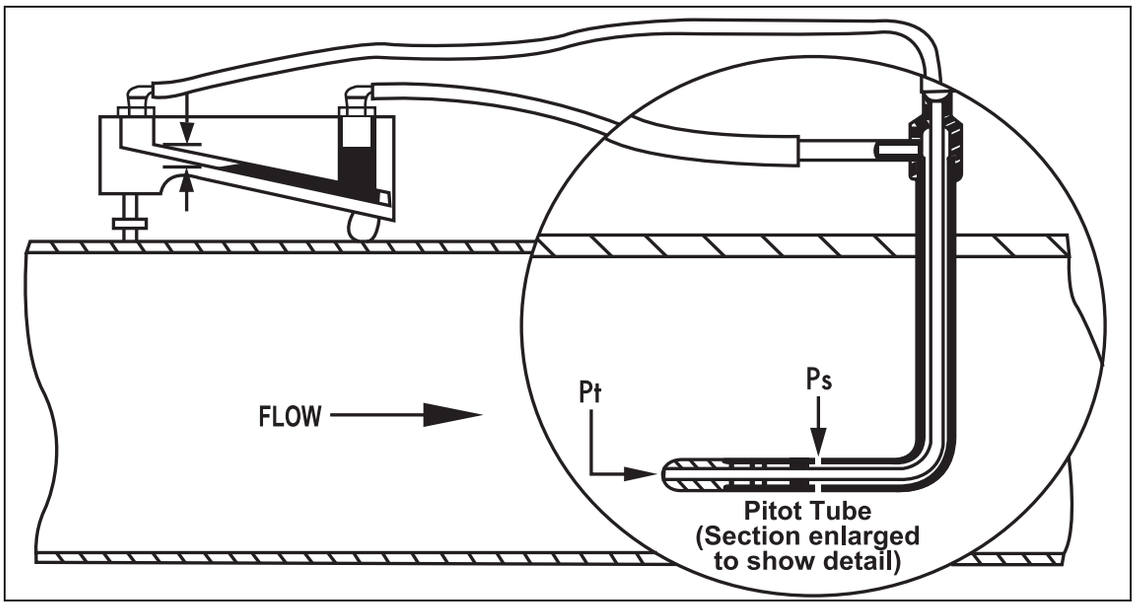


Figure 6.5 Pitot tube senses total and static pressure. Manometer measures velocity pressure (Difference between total and static pressures)

Traverse readings: In practical situations, the velocity of the air stream is not uniform across the cross section of a duct. Friction slows the air moving close to the walls, so the velocity is greater in the center of the duct.

To obtain the average total velocity in ducts of 100 mm diameter or larger, a series of velocity pressure readings must be taken at points of equal area. A formal pattern of sensing points across the duct cross section is recommended. These are known as traverse readings. Figure 6.6 shows recommended Pitot tube locations for traversing round and rectangular ducts.

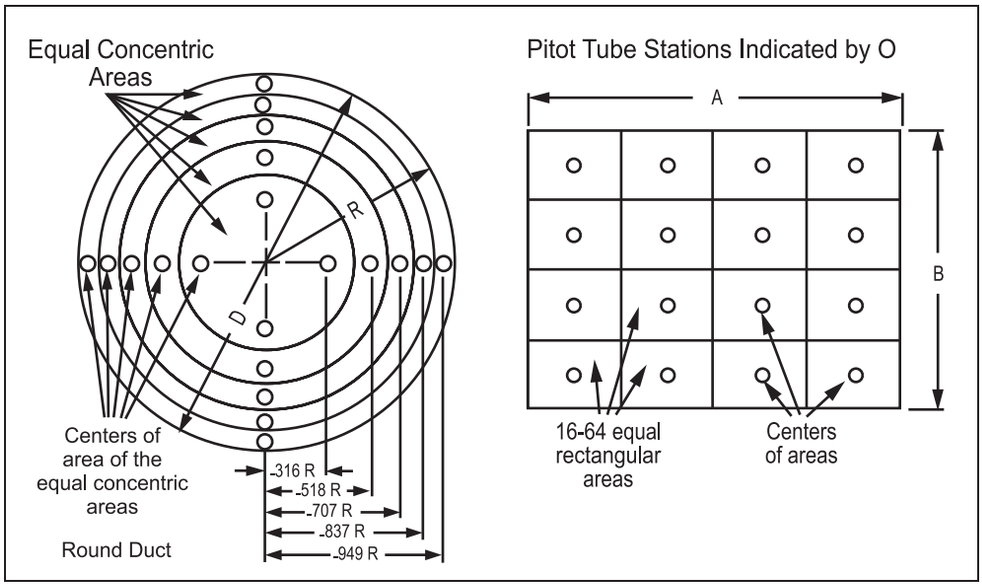


Figure 6.6 Traverse on Round and Square Duct Areas

In round ducts, velocity pressure readings should be taken at centers of equal concentric areas. Numbers of points or traverse points depend upon the diameter of duct. For a circular duct of 1 m diameter, atleast 10 traverse points are recommended. In rectangular ducts, a minimum of 16 and a maximum of 64 readings are taken at centers of equal rectangular areas.

Actual velocities for each area are calculated from individual velocity pressure readings. This allows the readings and velocities to be inspected for errors or inconsistencies. The velocities are then averaged.

Example-Traverse point determination for round duct

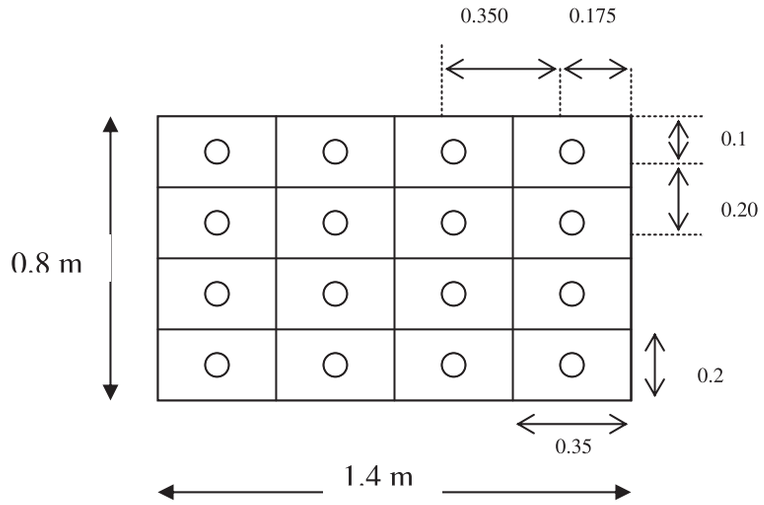
Round duct: Let us calculate various traverse points for a duct of 1 m diameter. From Figure 6.4, for round duct of 1 m diameter (D). The radius, R is 0.5 m. The various points from the port holes are given below:

0.5 - 0.949 x 0.5	0.0255
0.5 - 0.837 x 0.5	0.0815
0.5 - 0.707 x 0.5	0.1465
0.5 - 0.548 x 0.5	0.226
0.5 - 0.316 x 0.5	0.342
0.5 + 0.316 x 0.5	0.658
0.5 + 0.548 x 0.5	0.774
0.5 + 0.707 x 0.5	0.8535
0.5 + 0.837 x 0.5	0.9185
0.5 + 0.949 x 0.5	0.9745

Example-Traverse point determination for rectangular duct

Rectangular duct: For 1.4 m x 0.8 m rectangular duct, let us calculate the traverse points. 16 points are to be measured.

Dividing the area 1.4 x 0.8 = 1.12 m^2 into 16 equal areas, each area is 0.07 m^2. Taking dimensions of 0.35 m x 0.20 m per area, we can now mark the various points in the rectangular duct as follows:



Calculation of Velocity: After taking velocity pressures readings, at various traverse points, the velocity corresponding to each point is calculated using the following expression.

$$\text{Velocity (m/s)} = C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}}$$

Where,

C_p = The pitot tube coefficient (take manufacturer's value or assume 0.85)

ΔP = The average velocity pressure measured using pitot tube and inclined manometer at an identified traverse point across the cross section of the duct, mm WC.

γ = Gas density at flow conditions, kg/m^3 corrected to normal temperature.

The indicated velocity shall be measured at each traverse point in the cross section by holding the pitot tube stationary at each point. Velocity pressure reading shall be converted to velocity in m/s. The arithmetic mean of velocities at all traverse points gives the average velocity in the air duct.

Calculation of gas density corrected to normal temperature

$$Density\,(\gamma), kg/m^3 = \frac{P \times M}{R \times T}$$

Where,

P – Absolute gas pressure, mmWC
M – Molecular weight of the gas, kg/kg mole
T – Gas temperature, K
R – Gas constant, 847.84 mmWC m^3/kg mole K

Calculation of molecular weight of the gas (M) (dry basis), kg/kg mole

= {% CO_2 x M_{CO2} + % O_2 x M_{O2} + % CO x M_{CO} + % N_2 x M_{N2} + % SO_2 x Mso_2}/100

The indicated velocity shall be measured at each traverse point in the cross section by holding the anemometer stationary at each point for a period of time of not less than 1 minute. Each reading shall be converted to velocity in m/s and individually corrected in accordance with the anemometer calibration. The arithmetic mean of the corrected point velocities gives the average velocity in the air duct and the volume flow rate is obtained by multiplying the area of the air duct by the average velocity.

6.6.4 Determination of Flow

Once the cross-sectional area of the duct is measured, the flow can be calculated as follows:

Flow, (m^3/s) = Area (m^2) x Velocity (m/s)

6.6.5 Determination of Fan Static Pressure

The measurements of the static pressure on the inlet and outlet sides of the fan are taken relative to the atmosphere pressure using manometer in conjunction with the static pressure connection of a pitot tube or a U tube manometer.

6.6.6 Determination of Power Input

Power Measurement: The power measurements can be done using a suitable clamp- on power meter. Alternatively by measuring the amps, voltage and assuming a power factor of 0.9 the power can be calculated as below:

$$P = \sqrt{3} \times V \times I \times \cos\Phi$$

Transmission Systems: If fan is not connected to motor directly, transmission efficiency should be suitable assumed depending upon the type.

Directly coupled	1
Properly lubricated precision spur gears	0.98 for each step
Flat belt drive	0.97
V-belt drive	0.95

Other Prime Movers: When the fan is connected to non-electric prime mover instead of motor, it is recommended that the fuel consumption (oil, steam, compressed air etc.) should be specified and determined in place of the overall power.

$$Power\ Input\ to\ fan\ shaft = Power\ input\ to\ the\ motor * \eta\ of\ motor\ at\ the\ corresponding\ loading * transmission\ system\ \eta$$

$$Static\ fan\ efficiency\ (\%) = \frac{Volume\ in\ m^3/\sec * Static\ pressure\ in\ mmWC}{102 * Power\ input\ to\ fan\ shaft\ in\ kW}$$

6.7 Factors that Could Affect Performance

- Leakage, re-circulation or other defects in the system;
- Inaccurate estimation of flow resistance;
- Erroneous application of the standardized test data;
- Excessive loss in a system component located too close to the fan outlet;
- Disturbance of the fan performance due to a bend or other system component located too close to the fan inlet;
- Error in site measurement

Solved Examples:

1) A cement kiln exhaust gas has the following composition on dry basis is CO_2 - 24.7%, O_2 – 5.1%, CO – 0.1%, N_2 – 70.1 %. The static pressure and temperature measured in the duct are -655 mmWC and 316 ^{0}C. The velocity pressure measured with a pitot tube is 16.39 mmWC. The atmospheric pressure at the site is 10334 mmWC. The pitot tube constant is 0.85. If the area of the duct is 8.3 m^2. Calculate the volumetric flow rate in m^3/hr.

Solution:

Calculation of molecular weight of the gas, M (dry basis)

= % CO_2 x M_{CO2} + % O_2 x M_{O2} + % CO x M_{CO} + % N_2 x M_{N2}
= (24.7 x 44) + (5.1 x 32) + (0.1 x 28) + (70.1 x 28)
= 32.16 kg/kg mole

Calculation of gas density at operating temperature

$$Density\,(\gamma), kg/m^3 = \frac{P \times M}{R \times T}$$

= (10334 - 655) x 32.16 / {847.84 x (273 +316)}
= 0.64 kg/m^3

Determination of Flow

$$\text{Volumetric flow rate, } m^3/sec = A \times C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}}$$

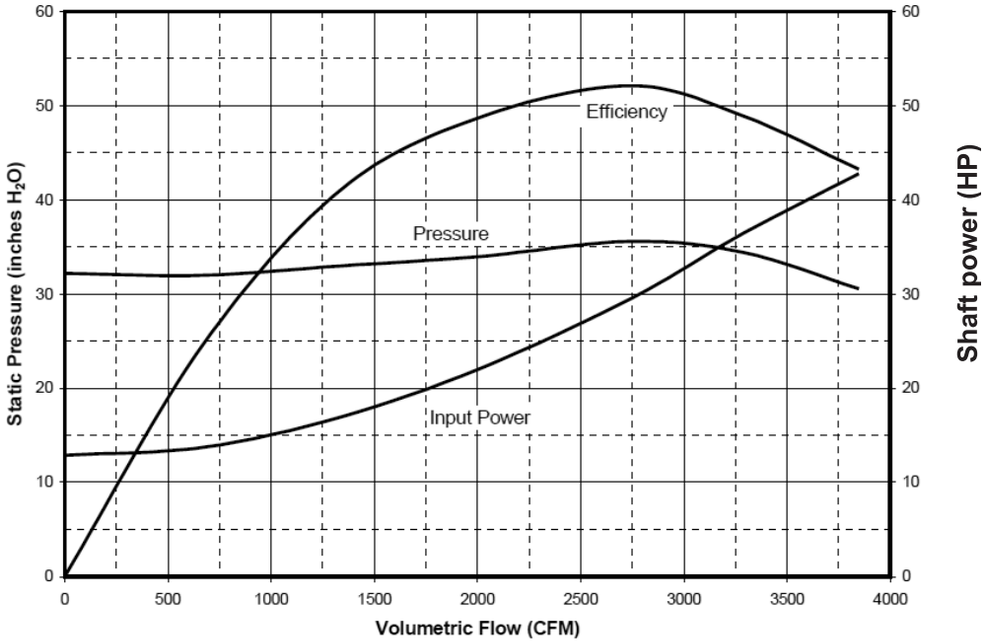
$$\text{Volumetric flow rate, } m^3/sec = 8.3 \times 0.85 \times \sqrt{\frac{2 \times 9.81 \times 16.39}{0.64}}$$

= 158.14 m^3/sec

= 5,69,304 m^3/hr

2) Given below is a set of curves for a centrifugal fan. At its Best Efficiency Point (BEP) determine to the nearest approximation the following:

a) Static pressure in mmwc
b) Flow in m^3/hr
c) Shaft power in kW
d) Work out the static efficiency of the fan by calculation
e) Power drawn by the motor if the motor operating efficiency is 90%



Solution:

Static pressure in inches H_2O	36
Static pressure in mmwc	914.4 mmwc
Flow in CFM	2750
Flow in m^3/hr	4672.28 m^3/hr ie 1.3 m^3/sec
Shaft power in hp	29
Shaft power in kW	21.6 kW
Fan Efficiency	Q x dP/(102 x kW)
	1.3 x 914.4 /(102 x 21.6)
	54 %
Motor input power at 90%	21.6 / 0.9 = 24 kW

3) The following is a typical report on measurements taken and calculations made for a *double* **inlet fan** in a palletizing plant**.**

A. Design Parameters:

Volume	= 292 m^3/sec.
Static Pressure	= 610 mmwc

B. Measurements:

Instruments used

a) Suction pressure, outlet pressure	= 'U' tube manometer
b) For differential pressure	= Inclined tube manometer
c) For temperature	= Mercury in glass thermometer
d) Fan speed	= Tachometer
e) Line current	= Tong tester

Ambient temperature	= 32°C
Atmospheric pressure	= 10334 mm WC
Area of duct (double entry)	= 1.029 x 5.502 x 2

Fan

Damper position	= 80% open
Speed	= 740 RPM
Average Static pressure (suction to fan)	= -20 mm WC
Average outlet pressure (discharge from fan)	= 470 mm WC

Average of velocities measured at various Traverse points	= 29 m/s

Velocity pressure or velocity head is measured at each traverse point and velocity is calculated. Average velocity is arithmetic mean of the above calculated velocities.

Motor

Current	= 220 A
Voltage	= 6.6 kV
PF	= 0.9
Motor power input = [√3 x KV x I x Cos Φ]	= 2263 (calculated) [√3 x 6.6 x 220 x 0.90]

Motor efficiency	= 0.94
Transmission efficiency	= 1 (directly coupled)
Power input to fan shaft	= 2127 kW

C. Performance calculations:

a) Calculation of gas density

$$Density\,(\gamma), kg/m^3 = \frac{P \times M}{R \times T}$$

P - Absolute static pressure, mm WC

M - Molecular weight of air 28.84 kg/kg mole

R - Gas constant 874.84 mmWC m^3/kg mole K

Since the velocity measurements were made on suction side, the average static pressure is -20 mmWC. The molecular weight of air is 28.84 kg/kg mole.

= {(10334 -20) x 28.84} / {847.84 x (273 +32)}

= 1.15 kg/m^3

b) Flow rate (one side duct) = Velocity x Area

= 29 x 1.029 x 5.502

= 164.3 m^3/s

Flow rate (both side ducts) = 164.3 x 2 = 328.6 m^3/s (both inlet ducts)

Total air flow rate = 328.6 m^3/s

(Sample calculation for velocity pressure (velocity head of 70 mmWC))

$$\text{Volumetric flow, } m^3/s = C\,x\,A\,x\,\frac{\sqrt{2\,x\,9.81\,x\,\Delta p\,x\,\gamma}}{\gamma}$$

C = Pitot tube coefficient, 0.85

A = Area of rectangular duct in m^2, 1.029 x 5.502

Δp = Velocity pressure measured by pitot tube at one of traverse points over the entire cross section of the duct.(say 70 mm WC)

γ = Density at test condition, 1.15 kg/m^3

$$\text{Flow, } m^3/s = \frac{0.85 \times 1.029 \times 5.502 \times \sqrt{2 \times 9.81 \times 70 \times 1.15}}{1.15}$$

= 166.3 m^3 / Sec

c) Static pressure developed by fan = 470 –(-20)

= 490 mmWC

d) Power input to the motor (kW) = 2263 kW

e) Power input to the fan shaft = 2127 kW

f) **Static Fan Efficiency (%)** $= \frac{\text{Flow, } m^3 / \text{Sec x } \Delta p \text{ Static pressure, mmwc}}{102 \text{ x Power input to the shaft in (kW)}}$

where 102 is a conversion constant

Static Fan Efficiency $= \frac{328.6 \text{ x } 490}{102 \text{ x } 2127} \text{ x } 100$

= 74%

Fan is designed to operate at 78% efficiency at the design point. However, fan is operating at lower flow than the design by partially closing the damper. Hence the fan is operating away from the best efficiency point.

QUESTIONS	
S-1	What is the relationship between static pressure, dynamic pressure and total pressure?
S-2	Mention the basic parameters that have to be measured during efficiency testing of a Fan.
S-3	Define dynamic Pressure?
S-4	How is velocity calculated using dynamic pressure?.
S-5	For determining fan efficiency, why static pressure readings should be taken as close to fan as possible?
S-6	What is the significance of having traverse points in velocity measurement?
S-7	The inclined manometer connected to a pitot tube is sensing which pressure in a gas stream?
S-8	Calculate the density of air at the measurement temperature of 40°C ignoring pressure correction.
S-9	Static fan efficiency = (m^3/s) x pressure gain in Pascal / Power input to shaft in Watt - Right or wrong?. Justify your answer.
S-10	How many traverse points you would propose for a rectangular duct of 1 m x 1 m dimensions?
L-1	Calculate the flow rate for the following data: Diameter of duct: 0.5 m, differential pressure: 100mmWC, static pressure: 10 mmWC, atmospheric pressure: 10000 mmWC, Temperature of air in the duct: 100°C, Pitot coefficient: 0.85
L-2	A direct coupled fan is drawing 28 kW. Its speed is brought down to 80% of the full speed. What will be the power drawn by the fan at this speed?
N-1	For a coal fired boiler an ID fan is used for controlling the draft. During the performance analysis of ID fan the following parameters are measured for analyses. Coal flow to the boiler = 15 TPH Flue gas density = 0.8 kg/m^3 ID fan – IGV% opening = 65% ID fan motor input power = 85 kW ID fan suction static pressure = -120 mmWC ID fan discharge static pressure = 15 mmWC Total flue gas quantity at APH inlet = 94 tph Flue gas temperature at ID fan inlet = 120°C O_2% before APH = 4.0% O_2% before ID fan = 8.0% **Calculate** (a) Infiltration air quantity (b) Fan static efficiency (consider motor efficiency as 90%) (c) Expected annual energy saving after arresting the infiltration for the following change in parameters.

REFERENCES

1. British Standard: BS 848 : Part 1 : 1980
2. Energy and Environmental Audit Reports of National Productivity Council

Chapter 7

Performance Evaluation of Pumps

Learning Objectives

In this chapter you will learn about

- Pump: Performance terms and definitions
- Different methods of assessing the flow
- How to determine the total head?
- How to estimate the efficiency of the pump?
- How to determine the system resistance and duty point?

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Estimate the actual flow rate of the pump using different methods**
- ✓ **Estimate pump efficiency**

7. ENERGY PERFORMANCE ASSESSMENT OF PUMPS

7.1 Introduction

Pumping is the process of addition of kinetic and potential energy to a liquid for the purpose of moving it from one point to another. This energy will cause the liquid to do work such as flow through a pipe or rise to a higher level. A centrifugal pump transforms mechanical energy from a rotating impeller into a kinetic and potential energy required by the system.

The most critical aspect of energy efficiency in a pumping system is matching of pumps to loads. Hence even if an efficient pump is selected, but if it is a mismatch to the system then the pump will operate at very poor efficiencies. In addition efficiency drop can also be expected over time due to deposits in the impellers. Performance assessment of pumps would reveal the existing operating efficiencies in order to take corrective action.

7.2 Purpose of the Performance Test

- Determination of the pump efficiency during the operating condition
- Determination of system resistance and the operating duty point of the pump and compare the same with design

7.3 Performance Terms and Definitions

Pump Capacity, Q = Volume of liquid delivered by pump per unit time, m^3/hr or m^3/sec

Q is proportional to N, where N- rotational speed of the pump

Total developed head, H = The difference of discharge and suction pressure

The pump head represents the net work done on unit weights of a liquid in passing from inlet of the pump to the discharge of the pump.

There are three heads in common use in pumps namely

(i) Static head
(ii) Velocity head
(iii) Friction head

The frictional head in a system of pipes, valves and fittings varies as a function (roughly as the square) of the capacity flow through the system.

System resistance: The sum of frictional head in resistance and total static head.

Pump Efficiency: Fluid power and useful work done by the pump divided by the power input in the pump shaft.

$$Pump\ Efficiency = \frac{Hydraulic\ power, P_h}{Power\ input\ to\ the\ pump\ shaft}\ x\ 100$$

Where,

Hydraulic power, P_h (kW) = Q x (h_d - h_s) x ρ x g / 1000

Q = Volume flow rate (m^3/s), ρ = density of the fluid (kg/m^3), g = acceleration due to gravity (m/s^2), (h_d - h_s) = Total head in metres

7.4 Field Testing for Determination of Pump Efficiency

To determine the pump efficiency, three key parameters are required: Flow, Head and Power. Of these, flow measurement is the most crucial parameter as normally online flow meters are hardly available, in a majority of pumping system. The following methods outlined below can be adopted to measure the flow depending on the availability and site conditions.

7.4.1 Flow Measurement, Q

The following are the methods for flow measurements:

- Tracer method BS5857
- Ultrasonic flow measurement
- Tank filling method
- Installation of an on-line flowmeter

Tracer Method

The Tracer method is particularly suitable for cooling water flow measurement because of their sensitivity and accuracy.

This method is based on injecting a tracer into the cooling water for a few minutes at an accurately measured constant rate. A series of samples is extracted from the system at a point where the tracer has become completely mixed with the cooling water. The mass flow rate is calculated from:

$$q_{cw} = q_{1\,x}\, C_1/C_2$$

Where, q_{cw} = cooling water mass flow rate, kg/s

q_1 = mass flow rate of injected tracer, kg/s
C_1 = concentration of injected tracer, kg/kg
C2 = concentration of tracer at downstream position during the 'plateau' period of constant concentration, kg/kg

The tracer normally used is sodium chloride.

Ultrasonic Flow meter

Operating under Doppler effect principle these meters are non-invasive, meaning measurements can be taken without disturbing the system. Scales and rust in the pipes are likely to impact the accuracy.

- Ensure measurements are taken in a sufficiently long length of pipe free from flow disturbance due to bends, tees and other fittings.
- The pipe section where measurement is to be taken should be hammered gently to enable scales and rusts to fall out.
- For better accuracy, a section of the pipe can be replaced with new pipe for flow measurements.

Tank filling method

In open flow systems such as water getting pumped to an overhead tank or a sump, the flow can be measured by noting the difference in tank levels for a specified period during which the outlet flow from the tank is stopped. The internal tank dimensions should be preferable taken from the design drawings, in the absence of which direct measurements may be resorted to.

Installation of an on-line flowmeter

If the application to be measured is going to be critical and periodic then the best option would be to install an on-line flowmeter which can get rid of the major problems encountered with other types.

7.4.2 Determination of total head, H

Suction head (h_s)

This is taken from the pump inlet pressure gauge readings and the value to be converted in to meters ($1kg/cm^2 = 10.$ m). If not the level difference between sump water level to the centerline of the pump is to be measured. This gives the suction head in meters.

Discharge head (h_d)

This is taken from the pump discharge side pressure gauge. Installation of the pressure gauge in the discharge side is a must, if not already available.

7.4.3 Determination of hydraulic power (Liquid horse power),

Hydraulic power, P_h (kW) = Q x (h_d - h_s) x ρ x g / 1000

Q = Volume flow rate (m^3/s), ρ = density of the fluid (kg/m^3), g = acceleration due to gravity (m/s^2), (h_d - h_s) = Total head in metres

7.4.4 Measurement of motor input power

The motor input power P_m can be measured by using a portable power analyser.

7.4.5 Pump shaft power

The pump shaft power P_s is calculated by multiplying the motor input power by motor efficiency at the existing loading.

$$P_s = P_m \times \eta_{Motor}$$

7.4.6 Pump efficiency

This is arrived at by dividing the hydraulic power by pump shaft power

$$\eta_{Pump} = \frac{P_h}{P_s}$$

Example of pump efficiency calculation

Illustration of calculation method outlined

A chemical plant operates a cooling water pump for process cooling and refrigeration applications. During the performance testing the following operating parameters were measured;

Measured Data

Pump flow, Q	0.40 m^3/ s
Power absorbed, P	325 kW
Suction head (Tower basin level), h_1	+1 m
Delivery head, h_2	55 m
Height of cooling tower	5 m
Motor efficiency	88 %
Type of drive	Direct coupled
Density of water	996 kg/ m^3

Pump efficiency

Flow delivered by the pump	0.40 m^3/s
Total head, h2 -(+h1)	54 m
Hydraulic power	0.40 x 54 x 996 x 9.81/1000 = 211 kW
Actual power consumption	325 kW
Overall system efficiency	(211 x 100) / 325 = 65 %
Pump efficiency	**65/0.88 = 74 %**

7.5 Determining the System resistance and Duty point

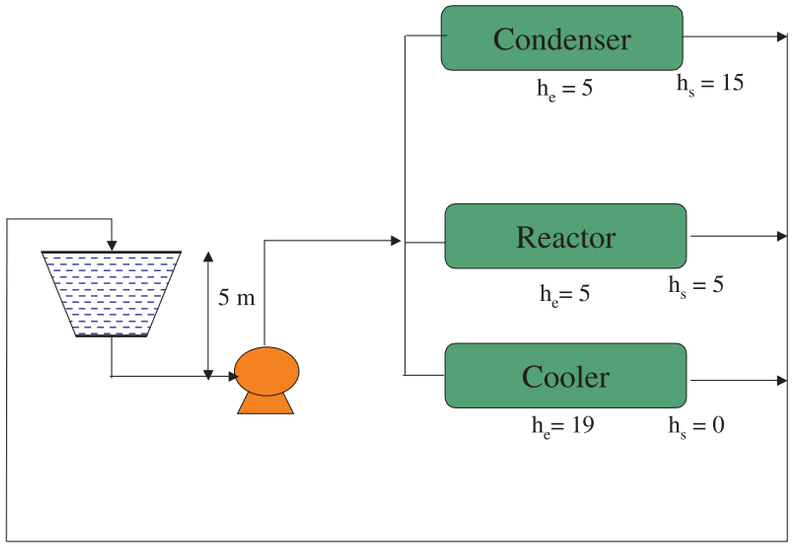
Determination of the system resistance curve and imposing the pump curve over it will give an idea of the operating efficiency of the pump and also the drop in efficiencies when the system curve changes from normal / design. The example following from the earlier example outlines the method of constructing a system curve.

Example:

Location of equipments

The Refrigeration plant is located at +0.00 level and the Process plant condensers are located at +15 m level. One cooler having a design pressure drop of 1.9 kg/cm^2 is located at the 0.00 level (ground level). Other relevant data can be inferred from the earlier section. See schematic in Figure 7.1.

Figure 7.1: Schematic of the System



h_s = Static Head ; h_e = Equipment pressure drop

The step-by-step approach for determining system resistance curve is given below.

Step-1 Divide system resistance into Static and dynamic head

Find static head;

Static head (Condenser floor height) = 15 m

Find dynamic head;

Dynamic Head = Total Head – Static Head

Dynamic head = (54-15) = 39 m

Step-2 Check the maximum resistance circuit

Resistance in the different circuits is as under

S.No.	System	Condenser loop resistance, m	Reactor loop resistance, m	Cooler loop resistance, m
1.	Supply line from pump	15	10	15
2.	Static head	15	5	Nil (cooler at ground level)
3.	Equipment	5	5	19
4.	Return line from equipment to CT	15	10	15
5.	Tower head	-	-	5
6.	Total	50	30	54

It can be noted that at full load the condenser and cooler circuits offer the maximum resistance to flow.

Step-3 Draw system resistance curve

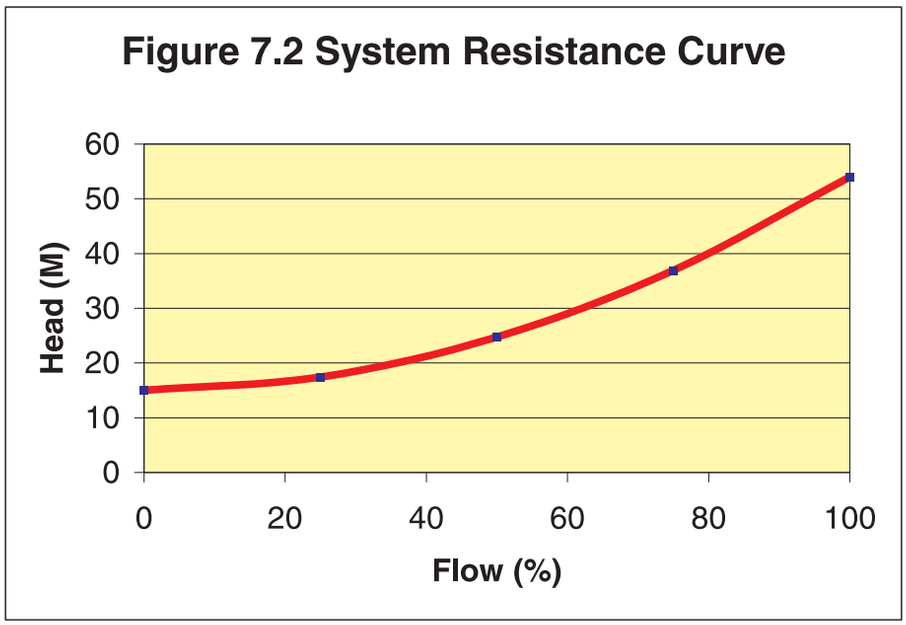
Choose the condenser loop as it offers maximum resistance and is also having a static head component

Static head: 15 m
Dynamic head at full load: 39 m

Compute system resistance at different flow rates

S.No.	Flow (%)	Dynamic head = 39x (%flow)^2	Static head, m	Total head, m
1.	100	39	15	54
2.	75	21.9	15	36.9
3.	50	9.75	15	24.75
4.	25	2.44	15	17.44

Step-4 Plot the system resistance against flow in the pump efficiency curves (see Figure 7.2) provided by the vendor and compare actual operating duty point and check whether it operates at maximum efficiency. In the example provided it is found that the pump system efficiency is lower by 4 % due to change in operating conditions.



Solved Examples:

1) In a municipality pumping system, water is pumped from the river to an underground circular sump of 8 metre dia in the intermediate booster station. Flow measurements were carried out by level difference in the sump. Pump takes 10 minutes to fill 1 metre level of circular sump. Pressure gauges are not available in the pumping system. The discharge pipe is horizontal, 300mm dia and 8 km long. Friction factor for the pipe is 0.006. The pump has a negative suction of 2 metre.

The details of power measurements at motor are:

3 phase voltage: 415 V, line current: 93 A and power factor: 0.89. The efficiency of the Motor is 0.90.

As an energy auditor, work out the following:

a) Flow rate of the pump in m^3/hr
b) Power drawn by the motor in kW
c) Total head developed by the pump (ignore friction losses in suction piping)
d) Operating efficiency of the pump

Solution:

Flow rate of the pump	$\frac{\prod \times 8^2 \times 60}{4 \quad \times 10}$
	302 m^3/hr
Power drawn by the motor	$\sqrt{3} \times v \times I \times \cos Ø$
	$\sqrt{3} \times 0.415 \times 93 \times 0.89$
	59.5 kW
Total head:	
Friction head	$4flv^2/2gD$
Velocity (flow/cross sectional area of pipe) metre/ second	$(302/3600)/(\prod \times 0.30^2/4)$
	1.19 m/s
Friction head	$\frac{4 \times 0.006 \times 1.19^2 \times 8000}{2 \times 9.81 \times 0.3}$
	46.2 metres
Total head developed by the pump	46.2 – (-2)
	48.2 m
Operating efficiency of the pump:	
Hydraulic power	(302/3600) x 48.2 x 9.81
	39.7 kW
Pump shaft power	59.5 x 0.9
	53.55 kW
Pump efficiency	39.7/53.55
	74.14 %

2) A centrifugal clear water pump rated for 800 m^3/hr was found to be operating at 576 m^3/hr with discharge valve throttled. The pumps speed is 1485 RPM. The discharge pressure of the pump before the throttle valve is 2 kg/cm^2g. The pump draws the water from a sump 4 metres below the centerline of the pump. The input power drawn by the motor is 124 kW at a motor efficiency of 92%.

(i) Find out the efficiency of the pump.

(ii) If the normal required water flow rate is 500 m^3/hr to 700 m^3/hr, what in your opinion should be the most energy efficient option to get the required flow rate variation?

(iii) And what would be the pump shaft power for that most energy efficient option if the pump is delivering the flow rate of 550 m^3/hr.

Solution:

(i) Hydraulic power, P_h (kW) = Q x (h_d - h_s) x r x g / 1000

Q = Volume flow rate (m^3/s), r = density of the fluid (kg/m^3), g = acceleration due to gravity (m/s^2), (h_d - h_s) = Total head in metres

h_d - h_s = 20 – (- 4) = 24 m

Hydraulic power = (576 x 24 x 1000 x 9.81) / (3600 x 1000)

= 37.67 kW

Input power to pump = 124 kW x 0.92 = 114 kW

Efficiency of the pump = (37.67 /114) x 100 = 33 %

(ii) Since the pump discharge requirement varies from 500 m^3/h to 700 m^3/h, the ideal option would be to operate with a VSD (variable frequency drive, hydraulic coupling)

(iii) For a flow rate 550 m^3/h, the reduced speed of pump would be:

(550/800) = (N1/1485)

N1 = 1021

The pump shaft power would be:

$$= \left(\frac{1021}{1485}\right)^3 \times 114 = 37 \text{ kW}$$

3) a) A centrifugal water pump operates at 30 m^3/hr and at 1440 RPM. The pump operating efficiency is 65% and motor efficiency is 89%. The discharge pressure gauge shows 3.4 kg/cm^2. The suction is 3 m below the pump centerline. If the speed of the pump is reduced by 25 %, estimate the following:

i) pump flow
ii) pump head and
iii) motor power

Assume motor and pump efficiency remains same at the reduced speed.

b) Calculate pressure drop in meters when pipe diameter is increased from 250 mm to 300 mm for a length of 600 meters. Water velocity is 2 m/s in the 250 mm diameter pipe and friction factor is 0.005.

Solution:

a)

Flow = 30 m^3/hr
Head developed by the pump = 34 – (-3) = 37 m
Power drawn by the pump = (30/3600) x 37 x 1000 x 9.81/(1000 x 0.65)
= 4.65 kW

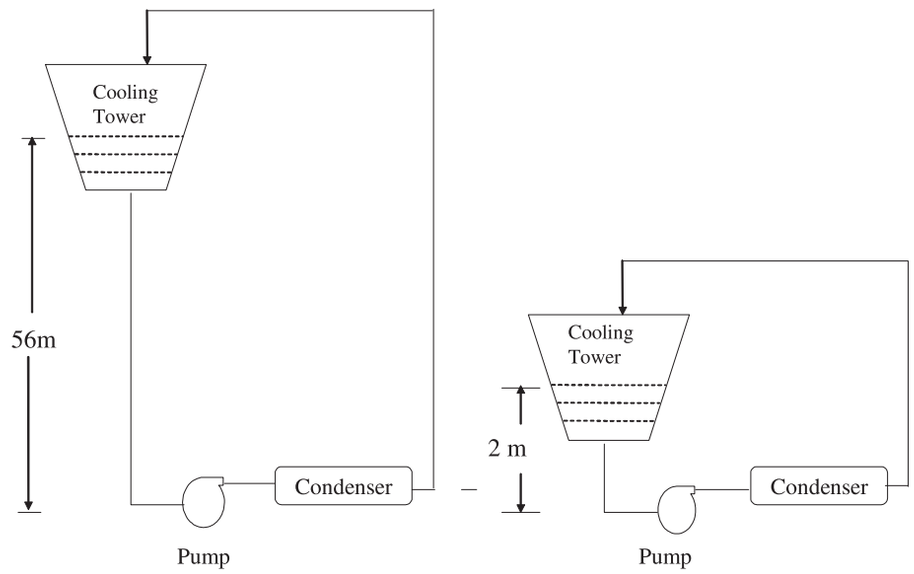
i) Flow at 75 % speed = 30 / Q2 = 1440/1080
= 22.5 m^3/hr
ii) Head at 75 % speed = 37 / H2 = $(1440/1080)^2$
= 20.81 m
iii) Shaft Power at 75 % speed = 4.65/kW2 = $(1440)^3$ / $(1080)^3$
= 1.96 kW
Power drawn by motor = 1.96 / 0.89
= 2.2 kW

b) Pressure drop = $\frac{4fLV^2}{2gD}$

Velocity of water in pipe of 300 mm diameter
= (0.25 x 0.25 x 2) / (0.3 x 0.3)
= 1.39 m/s

Pressure drop with 300 mm = 4 x 0.005 x 600 x 1.39^2 / (2 x 9.81 x 0.300)
= 3.94 m

4) In a commercial building, an energy auditor recommended to bring down the cooling tower from the terrace to the ground with a view to save energy in the pump. Details are given in the sketch below. Ignoring the friction losses, will this measure save energy? Explain with reason.



Answer:

No, because the pressure differential across the pump will be same as friction losses are ignored.

QUESTIONS	
S-1	How would you measure the flow by using tracer method?
S-2	Why does the friction head increase with increasing flow in a pumping system?
S-3	What is system resistance?
S-4	When should one consider impeller trimming option for a pump?
S-5	What parameters influence the hydraulic power in a pumping system?
S-6	A centrifugal pump raises water to a height of 12 metre. If the same pump handles brine with specific gravity of 1.2, to what height the brine will be raised?
S-7	Briefly explain static head and how it varies with flow rate?
S-8	State two methods of non-intrusive water flow measurements in a pipe.
S-9	The level in an overhead tank of 2m x 2m rises by 1m in 2 min. What is the flow rate of the pump?
S-10	By what percent the head will reduce if the pump's impeller is trimmed by 10%?
L-1	The suction head is 1m below the pump centerline. The discharge pressure shows 3 kg/cm^2. The flow is calculated to be 100 m^3/hr. Find out the pump efficiency.
L-2	The pump efficiency is 70%. The hydraulic power is calculated to be 22 kW. Find out the motor power required to drive the pump.
N-1	A centrifugal clear water pump rated for 800 m^3/hr was found to be operating at 576 m^3/hr with discharge valve throttled. The pumps speed is 1485 RPM. The discharge pressure of the pump before the throttle valve is 2 kg/cm^{2}g. The pump draws the water from a sump 4 metres below the centerline of the pump. The input power drawn by the motor is 124 kW at a motor efficiency of 92%. (i) Find out the efficiency of the pump. (ii) If the normal required water flow rate is 500 m^3/hr to 700 m^3/hr, what in your opinion should be the most energy efficient option to get the required flow rate variation? (iii) And what would be the pump shaft power for that most energy efficient option if the pump is delivering the flow rate of 550 m^3/hr.

REFERENCES

1. Pump handbook by Karassik
2. Energy Audit Reports of National Productivity Council

Chapter 8

Performance Assessment of Compressors

Learning Objectives

In this chapter you will learn about

- Terms used to define the performance of compressors
- How to find out the free air delivery of a compressor
- How to find out the efficiency of a compressor?
- How to find out the specific power consumption of the compressor?

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Measure actual free air delivered by the compressor**
- ✓ **Compute the isothermal power required and volumetric efficiency of the compressor**
- ✓ **Measure the specific power consumption of the compressor**

8. ENERGY PERFORMANCE ASSESSMENT OF COMPRESSORS

8.1 Introduction

The compressed air system is not only an energy intensive utility but also one of the least energy efficient. Over a period of time, both performance of compressors and compressed air system reduces drastically. The causes are many such as poor maintenance, wear and tear etc. All these lead to additional compressors installations leading to more inefficiency. A periodic performance assessment is essential to minimize the cost of compressed air.

8.2 Purpose of the Performance Test

To find out:

- Actual Free Air Delivery (FAD) of the compressor
- Isothermal power required
- Volumetric efficiency
- Specific power requirement

The actual performance of the plant is to be compared with design / standard values for assessing the plant energy efficiency.

8.3 Performance Terms and Definitions

Compression ratio	:	$\frac{\text{Absolute discharge pressure of last stage}}{\text{Absolute intake pressure}}$
Isothermal Power	:	It is the least power required to compress the air assuming isothermal conditions.
Isothermal Efficiency	:	The ratio of Isothermal power to Shaft power
Volumetric efficiency	:	The ratio of Free air delivered to Compressor swept volume
Specific power requirement:		The ratio of power consumption (in kW) to the volume delivered at ambient conditions.

8.4 Field Testing

8.4.1 Measurement of Free Air Delivery (FAD) by Nozzle method

Principle: If specially shaped nozzle discharge air to the atmosphere from a receiver getting its supply from a compressor, sonic flow conditions sets in at the nozzle throat for a particular

ratio of upstream pressure (receiver) to the downstream pressure (atmospheric) i.e. Mach number equals one.

When the pressure in the receiver is kept constant for reasonable intervals of time, the airflow output of the compressor is equal to that of the nozzle and can be calculated from the known characteristic of the nozzle.

8.4.2 Arrangement of test equipment

The arrangement of test equipment and measuring device shall confirm to Figure 8.1.

8.4.3 Nozzle Sizes

The following sizes of nozzles are recommended for the range of capacities indicated below:

Nozzle size (mm)	Capacity (m^3/hr)
6	3 – 9
10	9 –30
16	27 –90
22	60 – 170
33	130 – 375
50	300 – 450
80	750 – 2000
125	1800 – 5500
165	3500 - 10000

Flow Nozzle: Flow nozzle with profile as desired in IS 10431:1994 and dimensions

8.4.4 Measurements and duration of the test.

The compressor is started with the air from the receiver discharging to the atmosphere through the flow nozzle**. It should be ensured that the pressure drop through the throttle valve should be equal to or twice the pressure beyond the throttle.** After the system is stabilized the following measurements are carried out:

- Receiver pressure
- Pressure and temperature before the nozzle
- Pressure drop across the nozzle
- Speed of the compressor
- kW, kWh and amps drawn by the compressor

The above readings are taken for the 40%, 60%, 100% and 110% of discharge pressure values.

Measuring instruments required for test

- Thermometers or Thermocouple
- Pressure gauges or Manometers
- Differential pressure gauges or Manometers

- Standard Nozzle
- Psychrometer
- Tachometer/stroboscope
- Electrical demand analyser

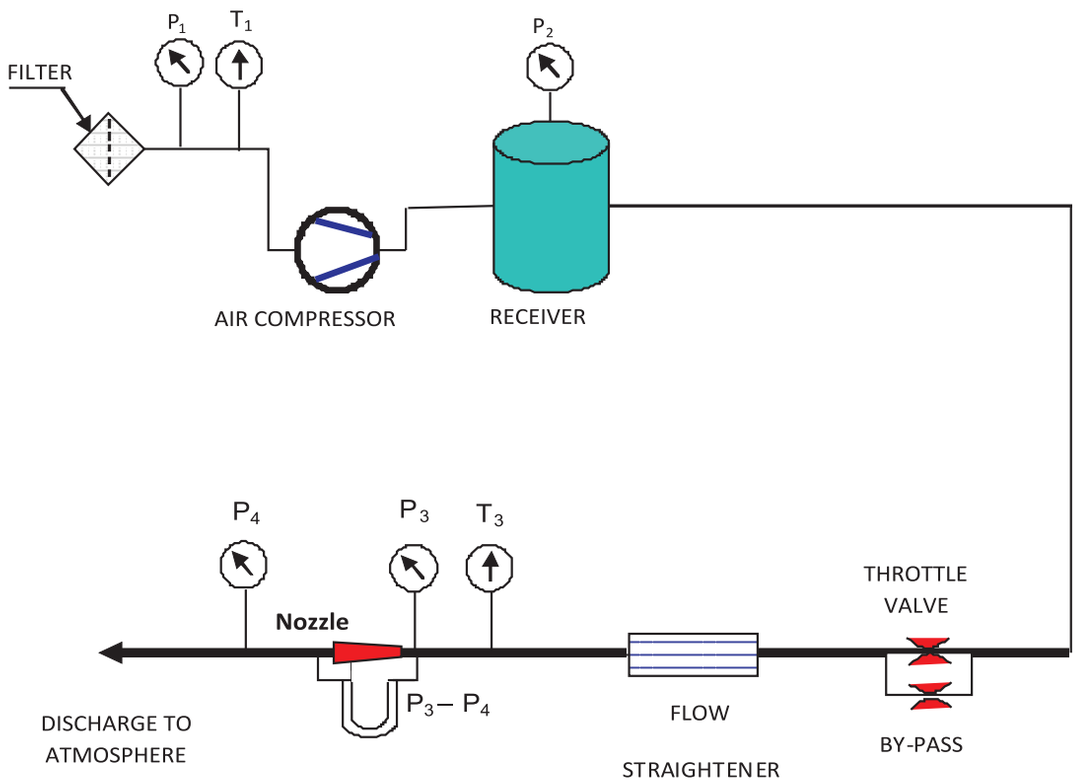


Figure 8.1: Test Arrangement for Measurement of Compressed Air Flow

8.5 Calculation Procedure for Nozzle Method

I. $$Free\,air\,delivered, Q_f (m^s / \sec) = k\,x\,\frac{\pi}{4}\,x\,d^2\,x\,\frac{T_1}{P_1}\,x\left(\frac{2(P_3 - P_4)(P_3\,x\,R_a)}{T_3}\right)^{1/2}$$

k	:	Flow coefficient – as per IS
d	:	Nozzle diameter M
T_1	:	Absolute inlet temperature ^oK
P_1	:	Absolute inlet pressure kg/cm^2
P_3	:	Absolute Pressure before nozzle kg/cm^2
T_3	:	Absolute temperature before nozzle ^oK
R_a	:	Gas constant for air 287.1 J/kg k
P_3-P_4	:	Differential pressure across the nozzle kg/cm^2

II. Isothermal Efficiency = Isothermal power / Input power

$$Isothermal\ power\ (kW) = \frac{P_1 \, x \, Q_f \, x \log_e r}{36.7}$$

P_1 = Absolute intake pressure kg/ cm^2
Q_f = Free air delivered m^3/hr.
r = Pressure ratio P_2/P_1

III. $$Specific\ power\ consumption\ at\ rated\ discharge\ pressure = \frac{Power\ consumption, kW}{Free\ air\ delivered, m^3/hr}$$

IV. $$Volumetric\ efficiency = \frac{Free\ air\ delivered\ in\ m^3/\min}{Compressor\ displacement\ in\ m^3/\min} \ x\ 100$$

$$Compressor\ displacement = \frac{\pi}{4} x\, D^2\, x\, L\, x\, S\, x\, \chi\, x\, n$$

D = Cylinder bore, metre
L = Cylinder stroke, metre
S = Compressor speed rpm
χ = 1 for single acting and
2 for double acting cylinders
n = No. of cylinders

8.6 Example

Calculation of Isothermal Efficiency for a Reciprocating Air Compressor.

Step – 1 : Calculate Volumetric Flow Rate

k : Flow coefficient (Assumed as 1)
d : Nozzle diameter : 0.08 metre
P_2 : Receiver Pressure - 3.5 kg / cm^2 (a)
P_1 : Inlet Pressure - 1.04 kg / cm^2(a)
T_1 : Inlet air temperature 30°C or 303°K
P_3 : Pressure before nozzle – 1.08 kg / cm^2
T_3 : Temperature before the nozzle 40°C or 313°K
$P_3 – P_4$: Pressure drop across the nozzle = 0.036 kg / cm^2
R_a : Gas constant : 287 Joules / kg K

$$Free\,air\,delivered, Q_f(m^s/\sec) = k\,x\,\frac{\pi}{4}\,x\,d^2\,x\,\frac{T_1}{P_1}\,x\left(\frac{2(P_3-P_4)(P_3\,x\,R_a)}{T_3}\right)^{1/2}$$

$$Free\,air\,delivered, Q_f(m^s/\sec) = 1\,x\,\frac{\pi}{4}\,x\,(0.08)^2\,x\,\frac{303}{1.04}\,x\left(\frac{2\,x\,0.036\,x\,1.08\,x\,287}{313}\right)^{1/2}$$

= 0.391 m^3/sec

= 1407.6 m^3 / h.

Step – 2 : Calculate Isothermal Power Requirement

$$Isothermal\ power\,(kW) = \frac{P_1\,x\,Q_f\,x\log_e r}{36.7}$$

P_1 -Absolute intake pressure = 1.04 kg / cm^2 (a)

Q_f -Free Air Delivered = 1407.6 m^3 / h.

Compression ratio, $r = \frac{3.51}{1.04} = 3.36$

$$Isothermal\ power\,(kW) = \frac{1.04\,x\,1407.6\,x\log_e 3.36}{36.7} = 48.34\,kW$$

Step – 3 : Calculate Isothermal Efficiency

Motor input power = 100 kW
Motor and drive efficiency = 86 %
Compressor input power = 86 kW

Isothermal efficiency = $\frac{\text{Isothermal Power x 100}}{\text{Compressor input Power}}$

= $\frac{\mathbf{48.34\ x\ 100}}{\mathbf{86.0}}$ = **56%**

8.7 Assessment of Specific Power requirement

Specific power consumption = $\frac{\text{Actual power consumed by the compressor}}{\text{Measured Free Air Delivery}}$

In the above example the measured flow is 1407.6 m^3/hr and actual power consumption is 100 kW.

Specific power requirement = $\frac{100}{1407.6}$

= 0.071 kW/m^3/hr

8.8 Measurement of FAD by Pump Up Method

(Note: The following section is a repeat of material provided in the chapter-3 on Compressed Air System in Book-3.)

Another way of determining the Free Air Delivery of the compressor is by Pump Up Method - also known as receiver filling method. Although this is less accurate, this can be adopted where the elaborate nozzle method is difficult to be deployed.

Simple method of Capacity Assessment in Shop floor

- Isolate the compressor along with its individual receiver being taken for test from main compressed air system by tightly closing the isolation valve or blanking it, thus closing the receiver outlet.
- Open water drain valve and drain out water fully and empty the receiver and the pipeline. Make sure that water trap line is tightly closed once again to start the test.
- Start the compressor and activate the stopwatch.
- Note the time taken to attain the normal operational pressure P_2 (in the receiver) from initial pressure P_1.
- Calculate the capacity as per the formulae given below:

Actual Free air discharge

$$Q = \frac{P_2 - P_1}{P_0} \times \frac{V}{T} \text{ Nm}^3 / \text{Minute}$$

Where,

P_2	=	Final pressure after filling (kg/cm^2 a)
P_1	=	Initial pressure (kg/cm^{2}a) after bleeding
P_0	=	Atmospheric Pressure (kg/cm^2 a)
V	=	Storage volume in m^3 which includes receiver, after cooler, and delivery piping
T	=	Time take to build up pressure to P_2 in minutes

The above equation is relevant where the compressed air temperature is same as the ambient air temperature, i.e., perfect isothermal compression. In case the actual compressed air temperature at discharge, say $t_2{}^0$C is higher than ambient air temperature say $t_1{}^0$C (as is usual case), the FAD is to be corrected by a factor $(273 + t_1) / (273 + t_2)$.

Example

An instrument air compressor capacity test gave the following results (assume the final compressed air temperature is same as the ambient temperature) – Comment?

Piston displacement	:	16.88 m^3/minute
Theoretical compressor capacity	:	14.75 m^3/minute @ 7 kg/cm^2
Compressor rated rpm 750	:	Motor rated rpm : 1445
Receiver Volume	:	7.79 m^3
Additional hold up volume, i.e., pipe / water cooler, etc., is	:	0.4974 m^3
Total volume	:	8.322 m^3
Initial pressure P_1	:	0.5 kg/cm^2
Final pressure P_2	:	7.03 kg/cm^2
Atmospheric pressure P_0	:	1.026 kg/cm^2 a
Compressor output m^3/minute	:	$\frac{(P_2 - P_1) \times \text{Total Volume}}{\text{Atm. Pressure} \times \text{Pumpup time}}$
	:	$\frac{(7.03 - 0.5) \times 8.322}{1.026 \times 4.021} = 13.17$ m^3/minute

Capacity shortfall with respect to 14.75 m^3/minute rating is 1.577 m^3/minute i.e., 10.69 %, which indicates compressor performance needs to be investigated further.

Solved Example:

In a medium sized engineering industry a 340 m^3/hr reciprocating compressor is operated to meet compressed air requirement at 7 bar. The compressor is in loaded condition for 80% of the time. The compressor draws 32 kW during load and 7 kW during unload cycle.

After arresting the system leakages the loading time of the compressor came down to 60%.

Calculate the annual energy savings at 6000 hours of operation per year.

Solution:

Average power consumption with 80% loading

= [0.8 x 32 + 0.2 x 7] = 27 kW

Average power consumption with 60% loading after leakage reduction

= [0.6 x 32 + 0.4 x 7] = 22 kW

Saving in electrical power = 5 KW

Yearly savings = 5 x 6000

= 30,000 kWH

QUESTIONS	
S-1	What is meant by Free Air Delivery?
S-2	Define the term isothermal efficiency and explain its significance.
S-3	Define the term volumetric efficiency and explain its significance.
S-4	How is specific power requirement calculated?
S-5	What is meant by compression ratio for air compressor?
S-6	What is the correction factor for actual free air discharge in a compressor capacity test if compressed air discharge temperature is 15 ^{0}C higher than ambient air at 40 ^{0}C.
S-7	What condition is to be maintained to create sonic flow at the nozzle throat in the test set up for compressor capacity determination by nozzle method ?
S-8	In a compressor leak test if the load time 3 minutes and unload time is 6 minutes find out the leakage in the system in percentage?
S-9	What parameters influence the isothermal power requirement of compressed air?
S-10	What are the acceptable pressure drops in mains header and that in distribution system in a compressed air network?
L-1	Describe the method of estimating flow by pump up method.
L-2	Isothermal efficiency of a compressor is 63 %. Intake pressure is 1.03 kg/cm^2(a). The rated flow is 340 m^3/hr. Find out the compressor delivery pressure, if the shaft power drawn is 25 kW.
N-1	An engineering industry has a compressor of capacity 1680 m^3/h in operation. Free air delivery test of the compressor was carried out by filling the receiver. The test and other data are given below. Receiver capacity : 10 m^3 Interconnecting pipe : 1 m^3 Initial pressure in receiver : 1.0 kg/cm^{2}a Inlet air pressure to compressor 1.0 kg/cm^{2}a Final pressure : 8.25 kg/cm^{2}a Time taken to fill the receiver : 3 minutes (180 seconds) Inlet air temperature : 30 ^{o}C Air temperature in the receiver : 40 ^{o}C Motor rpm (D1) : 1400 Motor pulley diameter (N1) : 300mm Compressor rpm (D2) : 700 rpm Compressor Pulley diameter (N2) : 600 mm Average duration of loading : 40 minutes in an hour Average duration of unloading : 20 minutes in an hour Power consumption during loading : 150 kW Power consumption during unloading : 45 kW Cost of energy : Rs. 5.00 per kWh

	A : What is the operating free air delivery of the compressor? B: Evaluate the cost of energy per day (24hrs operation). C: The Plant was interested in reducing the unloading time of the compressor by reducing the pulley diameter of the motor. Evaluate the speed of the compressor required for 10 minutes unloading and 50 minutes and accordingly evaluate the diameter of the pulley of the motor. D: Estimate the hourly power consumption, energy savings after replacement of the pulley and payback period. Consider the cost of pulley and belts is Rs 40,000 and operating hours of the compressor is 8000 in a year. (consider during loading the power consumption is 120 kW and during unloading the power consumption was 35 kW)

REFERENCES

1. IS 10431:1994: Measurement of airflow of compressors and exhausters by nozzles.

2. IS 5456:1985 code of practice for testing of positive displacement type air compressors and exhausters

3. Compressor performance – Aerodynamics for the user by M Theodore Gresh-Butterworth Heinemann.

Chapter 9

Performance Evaluation of HVAC Systems

Learning Objectives

In this chapter you will learn about

- Performance terms and definitions of HVAC systems
- How to estimate the net refrigeration capacity?
- How to evaluate the performance of VCR system?
- How to evaluate the performance of VAR system?
- How to evaluate the performance of air handling unit?

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Measure the net refrigerating capacity**
- ✓ **Measure the performance of a Vapour Compression refrigeration (VCR) plant**
- ✓ **Measure the performance of a Vapour Absorption refrigeration (VAR) plant**
- ✓ **Measure the performance of Air Handling Units (AHU's)**

9. ENERGY PERFORMANCE ASSESSMENT OF HVAC SYSTEMS

9.1 Introduction

Air conditioning and refrigeration consume significant amount of energy in buildings and in process industries. The energy consumed in air conditioning and refrigeration systems is sensitive to load changes, seasonal variations, operation and maintenance, ambient conditions etc. Hence the performance evaluation will have to take into account to the extent possible all these factors.

9.2 Purpose of the Performance Test

The purpose of performance assessment is to verify the performance of a refrigeration system by using field measurements. The test will measure net cooling capacity (tons of refrigeration) and energy requirements, at the actual operating conditions. The objective of the test is to estimate the energy consumption at actual load vis-à-vis design conditions.

9.3 Performance Terms and Definitions

Tons of refrigeration (TR): One ton of refrigeration is the amount of cooling obtained by one ton of ice melting in one day: 3024 kcal/h, 12,000 Btu/h or 3.516 thermal kW.

Net Refrigerating Capacity: A quantity defined as the mass flow rate of the evaporator water multiplied by the difference in enthalpy of water entering and leaving the cooler, expressed in kcal/h, tons of Refrigeration. It means the actual operating TR.

kW/ton rating: Commonly referred to as efficiency, but actually power input to compressor motor divided by tons of cooling produced, or kilowatts per ton (kW/ton). Lower kW/ton indicates higher efficiency.

Coefficient of Performance (COP): Chiller efficiency measured in W output (heating) divided by W input (electric power). COP refers to heating efficiency of an air conditioner (heat pump). For example if an air conditioner generates 5 kW of heat from one kW electrical input, its COP is said to be 5.0.

Energy Efficiency Ratio (EER): Performance of smaller chillers and rooftop units is frequently measured in EER rather than kW/ton. EER is calculated by dividing a chiller's cooling capacity (in watts) by its power input (in watts) at full-load conditions. This definition of EER has been adopted in BEE star labeling programme. For example if an air conditioner generates 4000 W (1.14 TR) of cooling from 1000 W electrical input, its EER is said to be 4.0. The higher the EER, the more efficient the unit.

EER is also defined as ratio of chiller's cooling capacity (in BTU/hr) to its power input (in watts) at full-load conditions. For example if an air conditioner generates 13648 BTU/hr (1.14 TR) of cooling from 1000 W electrical input, its EER is said to be 13.65.

9.4 Components of HVAC Systems

The various components of HVAC systems are

i) Chiller plant

a) Evaporator unit

b) Compressor unit (in vapour compression refrigeration system)
Absorber and Generator unit (in vapour absorption refrigeration system)

c) Condenser unit (air cooled / water cooled and condenser fans / cooling tower fans)

ii) Air handling unit (with fans / blowers)

iii) Primary and secondary chilled/cooling water pumps

9.5 Procedure for Performance Evaluation of Vapour Compression Refrigeration (VCR) System

After establishing the steady-state conditions, three sets of data shall be taken, preferably at a minimum of five-minute interval. To minimize the effects of transient conditions, test readings should be taken simultaneously to the extent possible in all components of HVAC system.

9.5.1 To determine the net refrigeration capacity at the evaporator

The test shall include a measurement of the net heat removed from the water as it passes through the evaporator by measurement of the following:

a. Chilled water flow rate through the evaporator coil

b. Chilled water inlet / entering and outlet / leaving temperatures

The heat removed from the chilled water is equal to the product of the chilled water flow rate, the water temperature difference, and the specific heat of the water.

The net refrigeration capacity in tons shall be obtained by the following equation:

$$\text{Net Refrigeration Capacity (TR)} = \frac{Me \times C_p \times (t_{in} - t_{out})}{3024}$$

Where,
- Me - mass flow rate of chilled water, kg/hr
- C_p - Specific heat of water, kcal/kg ^oC
- t_{in} - Chilled water temperature at evaporator inlet, ^oC
- t_{out} - Chilled water temperature at evaporator outlet, ^oC

The accurate temperature measurement is very vital in refrigeration and air conditioning and least count should be at least one decimal.

Methods of measuring the water flow

In the absence of an on-line flow meter the chilled water flow can be measured by the following methods

- In case where hot well and cold well are available, the flow can be measured from the tank level dip or rise by switching off the secondary pump.
- Non invasive method would require a well calibrated ultrasonic flow meter using which the flow can be measured without disturbing the system
- If the waterside pressure drops are close to the design values, it can be assumed that the water flow of pump is same as the design rated flow.

9.5.2 Measurement of compressor power

The compressor power can be measured by a portable power analyser which would give reading directly in kW.

If not, the ampere has to be measured by the available on-line ammeter or by using a tong tester. The power can then be calculated by multiplying ampere with voltage and power factor.

$$\text{Power (kW)} = \sqrt{3} \times V \times I \times \cos\phi$$

9.5.3 To determine the heat rejected at the condenser

Heat rejected at condenser = Cooling load + Work done by compressor

$$\text{Heat Rejected (TR)} = \left(\text{Evaporator TR}\right) + \frac{\text{kW}}{3.516}$$

The shaft power kW absorbed (work done) by the compressor can be derived by measuring the motor input power multiplied by motor operating efficiency.

Heat rejected at the condenser can be measured as under:

Water cooled condenser :

a. Measure the water quantity flowing through the condenser using flow meter.

b. Measure the inlet and outlet temperature of water in the condenser using digital thermometer.

$$\text{Heat rejected (TR)} = \frac{\text{Mc x } C_p \times (t_{wo} - t_{wi})}{3024}$$

Where,

Mc - Mass flow rate of cooling water, kg/h
Cp - Specific heat of water, kcal/kg ^{o}C
t_{wi} - Cooling water temperature at condenser inlet, ^{o}C
t_{wo} - Cooling water temperature at condenser outlet, ^{o}C

Air cooled condenser :

a. Measure the air quantity flowing across condenser coil.

b. Measure the inlet and outlet temperatures of air using digital thermometer.

Since this process is normally a sensible heating, the capacity can be established by calculating only sensible heat gain.

$$\text{Heat rejected (TR)} = \frac{\text{Ma} \times C_{pa} \times (t_{ao} - t_{ai})}{3024}$$

Where,

Ma - Mass flow rate of air, kg/h
C_{pa} - Specific heat of air, kcal/kg ^{o}C
t_{ai} - Temperature of cooling air at condenser inlet ^{o}C
t_{ao} - Temperature of cooling air at condenser outlet ^{o}C

Measurement of air flow

Air flow may be measured with any of the following instruments:

a) Vane Anemometer
b) Hot wire anemometer

The measuring instruments should be duly calibrated. The least count for anemometers should be 0.1 m/s. Air flow rate is calculated as the multiplication product of the average air velocity in the plane of measurement and the flow area.

9.5.4 Performance calculations

The energy efficiency of a chiller is commonly expressed in one of the three following ratios:

1.Coefficient of performance		
COP	=	**kW refrigeration effect** / kW input
2. Energy efficiency ratio		
EER	=	**Watt refrigeration effect** / Watt input
3. Power per Ton		
kW/Ton	=	**kW input** / Tons refrigeration effect

First calculate the kW/ton rating from the measured parameters.

From this parameter, the other energy efficiency parameters can be calculated. It may be observed that the COP and EER values will be numerically the same. EER is used as a performance indicator only for window, split and package air conditioners.

9.5.5 Performance evaluation of air conditioning systems

For centralized air conditioning systems the air flow at the air handling unit (AHU) can be measured with an anemometer. The dry bulb and wet bulb temperatures can be measured at the AHU inlet and outlet. The data can be used along with a psychrometric chart (Figure 9.1) to determine the enthalpy (heat content of air at the AHU inlet and outlet)

$$Heat\ load\ (TR) = \frac{m \times (h_{in} - h_{out})}{4.18 \times 3024}$$

Where,

m – mass flow rate of air, kg/hr
h_{in} – enthalpy of inlet air at AHU, kJ/kg
h_{out} – enthalpy of outlet air at AHU, kJ/kg

Heat load can also be calculated theoretically by estimating the various heat loads, both sensible and latent, in the air-conditioned room (refer standard air conditioning handbooks). The difference between these two indicates the losses by way of leakages, unwanted loads, heat ingress etc.

9.5.6 Evaluation of fans / blowers

The following readings can be taken for evaluation.

a) Air flow rate
b) Static pressure developed by the fan pitot tube and manometer can be used for measuring the differential head.
c) RPM of the fan using tachometer / stroboscope.
d) Current (amps), Voltage, power factor and power consumed (kW) by the fan motor using power analyser.

The above readings establish the fan performance and can be compared with the design parameter.

9.5.7 Evaluation of primary and secondary water pumps

The following readings can be taken for evaluation.

a) Water flow rate
b) Head pressure developed. (measured with the help of suction and discharge pressure gauge)
c) RPM
d) Current (amps), Voltage, Power drawn (kW) by the motor.

The above readings can be compared with the performance chart or design parameter.

9.5.8 Performance assessment of Vapour Compression Refrigeration (VCR) system: Examples

Example 1: The following are the data collected during the energy audit of chilled water system in a chemical plant. Find out the kW/TR and COP?

Measurements data:

Parameter	*Units*	*Measured Data*
Power drawn by compressor motor	kW	112
Motor efficiency	%	92
Compressor circuit –2 A loading	%	61
Compressor circuit –2 B loading	%	59
Primary pump power	kW	11.8
Secondary pump power	kW	2.5
Chilled water flow through the primary circuit	m^3/hr	165
Chilled Water temp. Inlet to evaporator	^oC	8.8
Chilled Water temp. Outlet of evaporator	^oC	7.2

Calculations:

$$\text{Net refrigeration Capacity (TR)} = \frac{Me \times C_p \times (t_{in} - t_{out})}{3024}$$

$$\text{Net refrigeration Capacity (TR)} = \frac{165000 \times 1 \times (8.8 - 7.2)}{3024}$$

= 87.3 TR

$$\text{kW/ton rating} = \frac{\text{Compressor power, kW}}{\text{Net refrigeration capacity (TR)}}$$

$$\text{kW/ton rating} = \frac{112 \times 0.92 \text{ kW}}{87.3 \text{ TR}} = 1.18\, kW/TR$$

$$\textbf{COP} = \frac{\textbf{kW refrigeration effect}}{\textbf{kW input}}$$

$$\text{Coefficient of performance (COP)} = \frac{87.3 \times 3024}{103 \times 860}$$

$$= \quad 2.98$$

Example 2: Estimation of Performance for an office air conditioning system working on Vapour Compression Chilling Package having direct expansion type air handling unit.

Parameter	Unit	Value
Air flow (m)	m^3/h	22237
Air density (ρ_{air})	kg/ m^3	1.29
Air dry bulb temperature at evaporator inlet	°C	22.6
Air wet bulb temperature at evaporator inlet	°C	20.6
Enthalpy of air at evaporator inlet (h_{in})	kJ/kg	59.85
Air dry bulb temperature at evaporator outlet	°C	17.9
Air wet bulb temperature at evaporator outlet	°C	16.2
Enthalpy of air at evaporator outlet (h_{out})	kJ/kg	45.82
Refrigeration Effect (Q_e)		$m \times \rho_{air} \times (h_{in} - h_{out})$
	kJ/h	402461
	W	111795
Refrigeration Effect (R)		$Q_e / 3.51 \times 3600$
	TR	31.85
Power input to motor (W)	kW	35
Motor efficiency (η_m)	%	91
Drive transmission efficiency (η_t)	%	100 (direct driven)
Estimated Compressor shaft power (Q_{in})	kW	$W \times \eta_m \times \eta_t$
	kW	31.9
	W	31900
Energy Efficiency Ratio (EER)	W/W	Q_e / Q_{in}
	W/W	111795/31900
	W/W	3.5
Specific power consumption (SPC)	kW/TR	$31.9 / 31.85$
	kW/TR	1.0

Inference: The performance parameters estimated above are actual values at operating conditions, which should be compared with the design values. If there is any deviation from the design specifications, analysis of the above energy equipment have to be carried out.

Example 3: An air handling unit was evaluated for its performance in a study and the following parameters were recorded. The air velocity was measured at different locations of the face area of 1.19 m^2 for AHU unit and its average velocity was calculated as 1.3 m/s. The dry and wet bulb temperatures of the air at the inlet of the unit are 29.4 and 24.4 ^oC, and at the outlet 24.4 and 19.2^oC, respectively. Using psychrometric chart, determine the enthalpy of the air at inlet and outlet conditions and calculate the heat load in TR.

Solution:

Air flow rate = (1.19 m^2 x 1.3 m/s) x 3600 = 5569.2 m^3/h

Assuming air density as 1.2 kg/ m^3

Mass flow rate of air = 5569.2 x 1.2 = 6683 kg/h

From psychrometric chart, (for corresponding dry and wet bulb temperatures of air)

Enthalpy of the inlet air = 73.34 kJ/kg

Enthalpy of the outlet air = 54.5 kJ/kg

Heat load (TR) = [6683 x (73.34 – 54.5)] / (4.18 x 3024)

= 10 TR.

9.6 Procedure for Performance Evaluation of Vapour Absorption Refrigeration (VAR) System

It should be ensured that the evaporator, condenser and generator are nearly at same operating conditions throughout the duration of the test. The vapour absorption system auxiliaries include chiller water pumps, condenser water pumps, cooling tower fans and performance assessment of these auxiliaries can be carried out in the same manner as applicable to vapour compression refrigeration system auxiliaries.

9.6.1 Estimation of performance at evaporator side

The performance evaluation involves the measurement of following parameters.

Refrigeration effect (Q_e)

- Chilled water flow rate in the evaporator.
- Chilled water temperatures at the evaporator inlet and outlet.

Thermal power input (Q_{in})

- Steam mass flow rate in case of steam heated vapour absorption chilling package.
- Fuel flow rate in case of direct fuel fired vapour absorption package.

Measuring instruments:

The measuring instruments should be duly calibrated.

- Direct reading thermometers can be used for measuring temperature. The least count for temperature indicating instruments should be 0.1°C.
- The method mentioned in VCR system can be used for water flow rate measurement.
- For steam heated vapour absorption chilling package, the thermal power consumption may be measured with any of the following instruments:
  a. Calibrated in-line steam flow meter.
  b. Collection of condensate in calibrated volume (container) for a defined time period. The time period should be measured with a digital chronometer (stop-watch) with least count of 1/100 second. The condensate may be cooled to reduce the flash steam losses.
- For fuel fired vapour absorption systems, the thermal power may be measured with any of the following instruments:
  a. Calibrated In-line fuel flow meter.
  b. Fuel level difference (for liquid fuels) for a defined time period in a calibrated day tank. The time period should be measured with a digital chronometer (stop-watch) with least count of 1/100 second.

Performance calculations:

Coefficient of performance, COP at evaporator side

$$COP = \frac{Net\ refrigeration\ effect, Q_e}{Thermal\ power\ input, Q_{in}}$$

$$= \frac{M_e \times C_p \times (t_{in} - t_{out})}{3600 \times Q_{in}}$$

For Steam heated Vapour Absorption Chilling Packages,

$$Q_{in} = \frac{M_{st} \times (h_{st} - h_{cond})}{3600}$$

For Direct Fuel-Fired Vapour Absorption Chilling Packages,

$$Q_{in} = \frac{M_f \times GCV}{3600}$$

Where,

Q_{in} = Thermal energy input in kW
Q_e = Refrigeration effect in kW
M_e = Chilled water flow rate in the evaporator, kg/h
C_p = Specific heat of water, kJ/kg-K
t_{in} = Chilled water temperature at evaporator inlet, K
t_{out} = Chilled water temperature at evaporator outlet, K
M_{st} = Steam consumption rate, kg/hr
h_{st} = Enthalpy of steam at operating pressure, kJ/kg

h_{cond} = Enthalpy of condensate, kJ/kg
M_f = Fuel consumption rate, kg/hr
GCV = Gross calorific value of fuel, kJ/kg

9.6.2 Estimation of performance at condenser side (water cooled condenser)

The performance evaluation involves the measurement of following parameters.

Heat rejected at the condenser (Q_c)

- Cooling water flow rate in the condenser.
- Cooling water temperatures at absorber inlet and the condenser outlet.

Thermal energy input (Q_{in})

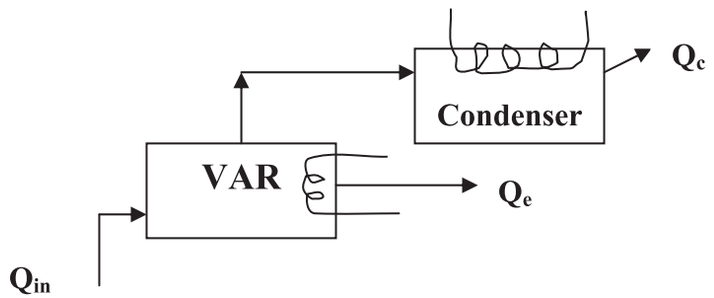
- Steam mass flow rate in case of steam heated vapour absorption chilling package.
- Fuel flow rate in case of direct fuel fired vapour absorption package.

Determination of refrigeration effect and COP:

i. The refrigeration effect and COP can be determined from the overall heat balance of the condenser and VAR system.
ii. Heat rejected at the condenser (Q_c) is equal to refrigeration effect heat (Q_e) plus thermal energy input (absorbed) by the system (Q_{in}).

Performance calculations:

a) COP at condenser side for steam heated vapour absorption package



At condenser,

Refrigeration effect heat = Heat rejected in the condenser – Thermal energy input

$$Q_e = Q_c - Q_{in}$$

$$COP = \frac{\text{Net refrigeration effect, } Q_e}{\text{Thermal energy input, } Q_{in}}$$

$$COP = \frac{Q_c - Q_{in}}{Q_{in}} = \frac{Q_c}{Q_{in}} - \frac{Q_{in}}{Q_{in}} = \frac{Q_c}{Q_{in}} - 1$$

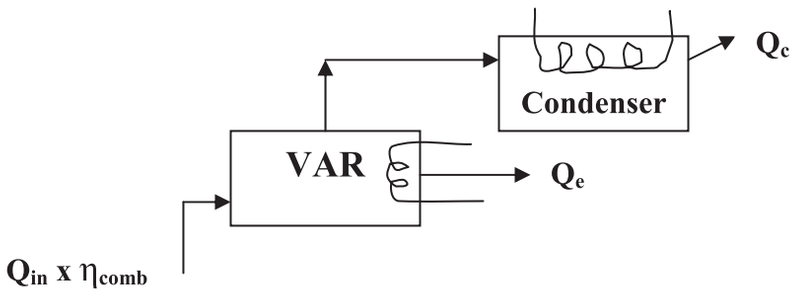
Therefore,

$$COP = \frac{\text{Heat rejected at condenser, } Q_c}{\text{Thermal energy input, } Q_{in}} - 1$$

$$= \frac{Mc \times Cp \times (t_{wo} - t_{wi})}{3600 \times Q_{in}} - 1$$

$$Q_{in} = \frac{M_{st} \times (h_{st} - h_{cond})}{3600}$$

b) COP at condenser side for direct fuel fired vapour absorption packages



At condenser,

Refrigeration effect heat = Heat rejected in the condenser – Heat input

$$Q_e = Q_c - Q_{in}\,\eta_{comb}$$

$$COP = \frac{Q_e}{Q_{in}}$$

$$COP = \frac{Q_c - Q_{in} \times \eta_{comb}}{Q_{in}} = \frac{Q_c}{Q_{in}} - \frac{Q_{in} \times \eta_{comb}}{Q_{in}} = \frac{Q_c}{Q_{in}} - \eta_{comb}$$

Therefore,

$$COP = \frac{\text{Heat rejected at condenser, } Q_c}{\text{Thermal energy input, } Q_{in}} - (\eta_{comb})$$

$$= \frac{M_c \times C_p \times (t_{wo} - t_{wi})}{3600 \times Q_{in}} - (\eta_{comb})$$

$$Q_{in} = \frac{M_f \times GCV}{3600}$$

Where,

Mc = Cooling water flow rate in the condenser, kg/h
t_{wi} = Cooling water temperature at absorber inlet, K for VAR chiller
t_{wo} = Cooling water temperature at condenser outlet, K
η_{comb} = Combustion efficiency

Combustion Efficiency Calculations:

The method described below can be used for estimating combustion efficiency of a direct fuel fired absorption chilling unit. The methodology is the "Indirect Method" to estimate combustion efficiency (η_{comb}), wherein the losses are estimated from flue gas analysis to estimate efficiency.

Observations:

Flue gas analysis: a) Average % O_2 (v/v)
b) Flue / stack gas temperature, T_f (°C)
c) Average ambient air temperature, T_a (°C)

Fuel analysis: a) % Moisture (w/w), M
b) % Carbon (w/w), C
c) % Hydrogen (w/w), H_2
d) % Oxygen (w/w), O_2
d) % Nitrogen (w/w), N_2
e) % Sulphur (w/w), S
f) Gross calorific valve, GCV (kcal/kg)

Calculations:

a) Theoretical air required for combustion	=	$[(11.6xC)+\{34.8x(H_2-O_2/8)\}+(4.35xS)]/100$ kg/kg of fuel. [from fuel analysis]
		Where C, H_2, O_2 and S are the percentage of carbon, hydrogen, oxygen and sulphur present in the fuel.
b) % Excess Air supplied (EA)	=	$\frac{O_2\%}{21-O_{2\%}} x100$ [from flue gas analysis]
c) Actual mass of air supplied/ kg of fuel (AAS)	=	{1 + EA/100} x Theoretical air

d) % Dry gas loss (D.G.L.):

D.G.L.	=	$\frac{mf \; x \; C_{pf} \; x \; (T_f - T_a)}{GCV \; of \; fuel} \; x \; 100$
Where,		
m_f	=	Mass of dry flue gas in kg/kg of fuel
Total mass of dry flue gas /kg of fuel = Actual mass of air supplied/kg of fuel +1 kg of fuel – ($9H_2$ + M)		
C_{pf}	=	Specific heat of flue gas in kcal/kg

e) % Wet gas loss (W.G.L.): $= \frac{(9 \text{ x } H_2 + M) \text{ x } \{584 + 0.45 \, (T_f - T_a)\}}{\text{GCV of fuel}} \; x \; 100$

Where, 584 = Latent heat corresponding to partial pressure of water vapour
0.45 = Specific heat of water vapour / superheated steam in kcal/kg°C

f) Total stack loss or flue gas loss (%) = D.G.L. + W.G.L.

g) Combustion efficiency by indirect method, η_{comb} = (100 - % Total flue gas losses) = Heat absorbed by the generator of VAR system.

Note: In this method only the major combustion losses namely, dry and wet flue gas losses are considered. All other losses are considered insignificant and hence ignored.

9.6.3 Performance assessment of Vapour Absorption Refrigeration (VAR) system: Example

This is a sample calculation for a Vapour Absorption Chilling Package. Measurements are shown along with equations and estimation of results.

Equipment Specification for VAR system

Parameter	Unit	Quantity
Rated Generator temperature (in case of VAR chilling package)		165
Rated Capacity at Full Load	TR	1180
Fluid being cooled in the evaporator		Water
Rated Evaporator Fluid Flow Rate, Me	m^3/h	713
Rated Evaporator inlet temperature, t_{in}	°C	12
Rated Evaporator outlet temperature, t_{out}	°C	7
Rated Condenser Water Flow Rate, Mc	m^3/h	1427
Rated Condenser inlet temperature, t_{wi}	°C	27
Rated Condenser outlet temperature, t_{wo}	°C	32

Method 1: Evaporator side

Estimation of performance from refrigeration effect in evaporator for steam heated vapour chilling packages (Chilling water)

Parameter	Unit	Value
Ambient dry bulb temperature	°C	38
Ambient wet bulb temperature	°C	25
Chilled water flow, M_e	m 3/h	620
Chilled water density, ρ	kg/ m^3	1000
Specific heat of chilled water, C_p	kJ/kg/K	4.18
Chilled water temperature at evaporator inlet, t_{in}	°C	8.8
Chilled water temperature at evaporator outlet, t_{out}	°C	6.8
Cooling Water flow, Mc	m^3/h	954
Cooling water inlet temperature, t_{wi}	°C	26.6
Cooling water outlet temperature, t_{wo}	°C	29.9
Refrigeration Effect, Q_e		$M_e \times \rho \times C_p (t_{in} - t_{out})$
	kJ/h	5183200
Refrigeration Effect, R		Qe / (3.516 x 3600)
	TR	410
Generator temperature	°C	120
Steam Pressure	kPa	300
Enthalpy of steam, h_{st}	kJ/kg	2734
Steam consumption rate, M_{st}	kg/h	3443
Condensate temperature	°C	100
Enthalpy of condensate, h_{cond}	kJ/kg	418
Estimated Thermal energy Input, Q_{in}	kJ/h	M_{st} X ($h_{st} - h_{cond}$)
	kJ/h	7974332
Coefficient of Performance, COP	kW/kW	Q_e / Q_{in}
		0.65
Specific Energy consumption, SEC	kJ/TR	M_{st}X (h_{st}-h_{cond})/R
	kJ/TR	3443 X (2734 – 418) / 410
	kJ/TR	19449
	kcal/TR	4645

Method 2: Condenser side

Estimation of performance from heat rejection in Water Cooled Condenser for steam heated vapour absorption chilling packages (Cooling water)

Parameter	Formula	Unit	Value
Heat Rejection, Qc	M_c x ρ x Cp x ($t_{wo} - t_{wi}$)	kJ/h	13159476
Refrigeration Effect, Qe	$Q_c - Q_{in}$	kJ/h	5185144
Refrigeration Effect, R	Q_e/ (3.516 x 3600)	TR	410
Coefficient of Performance, COP	Q_e / Q_{in}	kW/kW	0.65

Inference: The performance parameters estimated above show a deviation from the actual design values. This may due to operation of the chiller at less than 50% load.

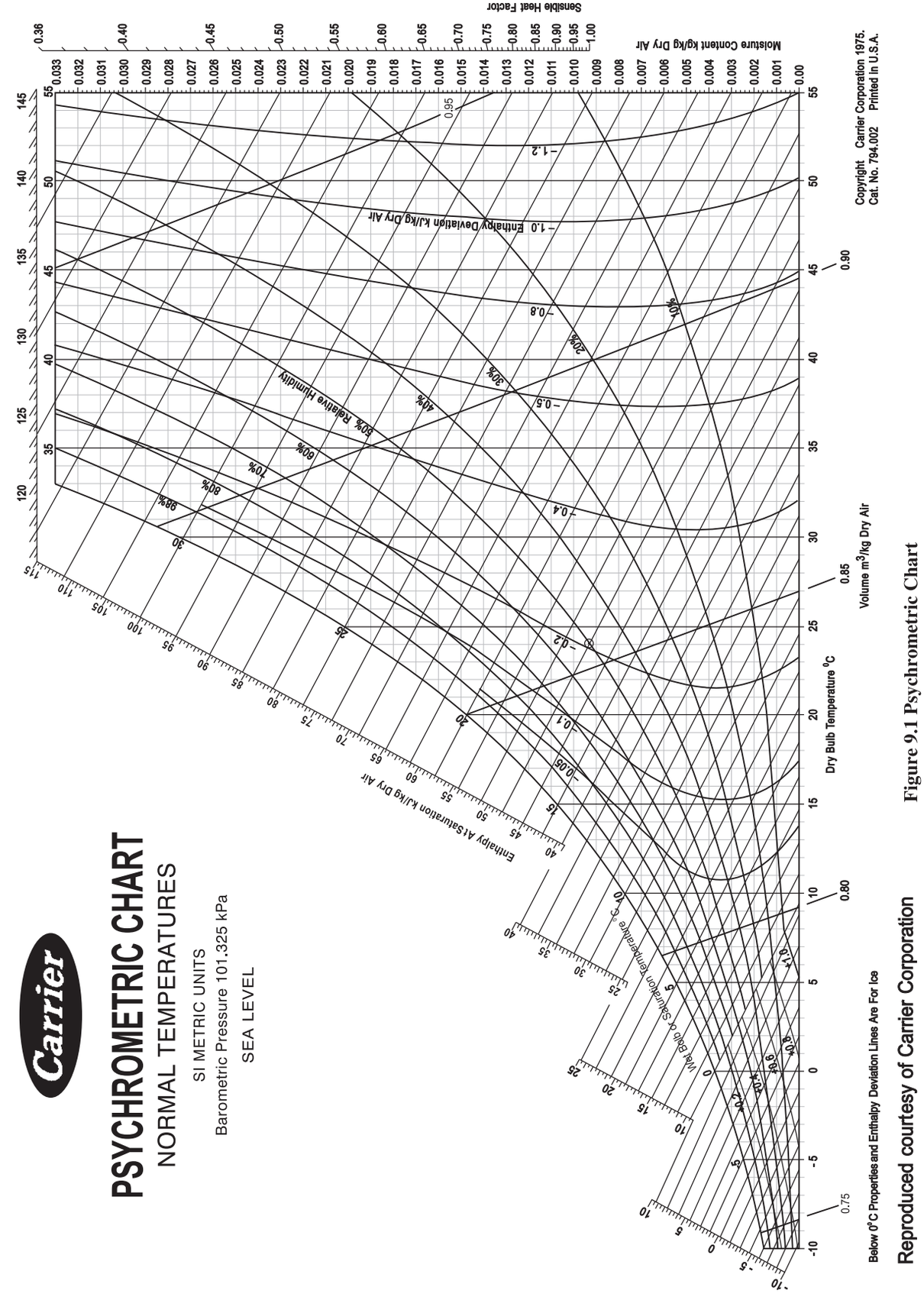


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Figure 9.1 Psychrometric Chart

Solved Example:

In an air conditioning system of a food processing industry, the cold air flow rate is 20,000 m^3/hr at a density of 1.2 kg/m^3 .The inlet and outlet enthalpy of the air are 105 kJ/kg and 80 kJ/kg. The COP of the existing vapour compression system is 3.75. The efficiency of the motor coupled with the compressor is 90%.

The management wants to install a Vapour Absorption System (VAR).The saturated steam for VAR will be supplied either from a new waste heat boiler to be installed with the existing DG sets or from the existing FO fuel fired boiler. The plant is operating for 8000 hr/annum. The investment of VAR system is Rs. 20 lakhs. The investment for waste heat boiler is Rs. 6 lakhs. The power cost is Rs. 6/ kWh.

As an energy auditor which one of the following options will you recommend to the management?

Option-1: Supply steam from the existing FO fuel fired boiler to VAR system and avoid the investment of waste heat boiler

Option-2: Supply steam from the waste heat boiler, which needs an investment in addition to VAR system

The steam consumption per TR will be 5.5 kg/TR. The cost of FO is Rs.32,000/ tonne. The evaporation ratio of the existing FO fired boiler is 14. Neglect losses in transmission of steam and chilled water.

Solution:

Existing Base Case VCR System	
TR Rating	$= \frac{(20{,}000\ m^3/hr \times 1.2\ kg/m^3)\ (105\text{-}80)\ kJ/kg}{3024 \times 4.187}$
	= 47.38 TR
COP	= 3.75
COP	$= \frac{\text{Refrigeration effect, kcal/hr}}{\text{Power Input, kcal/hr}}$
Compressor power input	$= \frac{47.38 \times 3024\ kcal}{3.75 \times 860}$
	= 44.43 kW
Motor input power	= 44.43/0.9 = 49.37 kW
Annual Energy Consumption	= 49.37 x 8000 = 3.95 Lakhs kWh
Annual cost in VCR system (Base Case)	=3.95 x 6 = Rs. 23.7 Lakhs

Option – 1 : **VAR System with Steam Supply from Existing Boiler**	
Steam Consumption /TR	= 5.5 kg/TR
Steam Consumption per hr	= 5.5 x 47.38 = 261 kg/hr
Evaporation Ratio IN THE EXISTING BOILER	= 14
1 ton of steam cost	= Rs.32000 / 14 = Rs.2.29 /kg of steam
Investment for VAR system	= Rs.20.00 Lakhs
Electricity cost saving per hr	= 49.37 x 6 =Rs. 296.22
Steam cost per hr	= 261 x 2.29 = Rs. 598
Since the steam cost per hour is higher than electricity cost this option is not feasible	
Option-2: With VAR & steam supply from WHR steam boiler of DG set	
Total Investment	= Rs. 20.00 + 6.00 = 26.00 Lakhs
Annual Savings	= Rs.23.7 lakhs
Simple Payback period	= 26/23.7 = 1.09 years.

Solution : Option 2 should be selected

QUESTIONS	
S-1	What is meant by a ton of refrigeration?
S-2	Is there a conceptual difference between COP and EER? Explain how they relate.
S-3	What is the relation between COP and kW/ton of refrigeration?
S-4	List the basic parameters required to assess the refrigeration capacity of chilled water system?
S-5	If the power consumed by a refrigeration compressor is 2 kW per ton of refrigeration, what is the energy efficiency ratio?
S-6	In an air conditioning system analysis which one temperature is sufficient to determine the enthalpy of air?
S-7	In the design of cooling tower for refrigeration system, besides refrigeration heat load which other heat load should be considered.
S-8	Explain why COP of VAR is much less than COP of vapour compression system.
S-9	If the dry bulb temperature of air is 35 ^{o}C and the wet bulb temperature is 35 ^{o}C what will be the relative humidity %.
S-10	In determining the heat load in the air conditioning system why enthalpy of air is considered rather than temperature?
L-1	List the energy saving opportunities in refrigeration systems?
L-2	How would you calculate the heat load for a room to be air-conditioned?
N-1	A pharma unit had installed a centralized refrigeration system of 120 TR Capacity several years ago. The refrigeration system operators 24 hours a day, 200 days per annum and the average electricity cost is Rs. 4.5/ kWh. The following are the key operational parameters. Compressor operating current and power factor : 153 amps. 0.9 pf Efficiency of the compressor motor : 0.92 Condenser pump operating current and power factor : 43 amps, 0.88 pf Chiller pump operating current and power factor : 25 amps, 0.9 pf CT fan operating current and power factor : 20 amps. 0.65 pf ΔT across the chiller (evaporator) : 3.5^{O}C Chilled water flow : 23 Lit / Sec Total head developed by chiller pump : 35 mtrs. Condenser water flow : 41 Lit / Sec Total head developed by condenser pump : 30 mtrs. PS: all the motors operate at 415 Volts.

	Calculate: • The power consumed by the compressor, condenser pump, chiller pump and CT fan. • TR developed by the system • Specific power consumption i.e. overall kW/TR and COP and Energy Efficiency ratio (EER) • Combined efficiency (motor and pump) of condenser and chiller pumps The unit proposes to replace the existing condenser and chilled water pumps with efficient pumps having a combined efficiency of 65%. Also the unit goes in for condenser cleaning by which the power consumption of compressor has reduced by 10%. Calculate: ▪ The envisaged power consumption of the compressor, condenser and chiller pump ▪ Hourly energy savings (compressor, condenser and chilled water pump) ▪ Annual energy and equivalent monetary savings (compressor, condenser and chilled water pump) ▪ Specific power consumption i.e. overall kW/ TR and COP and Energy Efficiency ratio (EER)

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Chapter 10

Performing Financial Analysis

Learning Objectives

In this chapter you will learn about

- Simple pay back period
- Return on investment
- Net present value
- Internal rate of return
- Sensitivity analysis

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Select and use appropriate financial analysis techniques for energy management project**
- ✓ **Carry out sensitivity analysis.**

10 PERFORMING FINANCIAL ANALYSIS

10.1 Introduction

When planning an energy efficiency or energy management project, the costs involved should always be considered. Therefore, as with any other type of investment, energy management proposals should show the likely return on any capital that is invested. Consider the case of an energy auditor who advises the senior management of an organisation that capital should be invested in new boiler plant. Inevitably, the management of the organisation would ask:

- How much will the proposal cost?
- How much money will be saved by the proposal?

These are, of course, not unreasonable questions, since within any organisation there are many worthy causes, each of which requires funding and it is the job of senior management to invest in capital where it is going to obtain the greatest return. In order to make a decision about any course of action, management needs to be able to appraise all the costs involved in a project and determine the potential returns.

This however, is not quite as simple as it might first appear. The capital value of plant or equipment usually decreases with time and it often requires more maintenance as it gets older. If money is borrowed from a bank to finance a project, then interest will have to be paid on the loan. Inflation too will influence the value of any future energy savings that might be achieved. It is therefore important that the cost appraisal process allows for all these factors, with the aim of determining which investments should be undertaken, and of optimising the benefits achieved. To this end a number of accounting and financial appraisal techniques have been developed which help energy managers and auditors make correct and objective decisions.

The financial issues associated with capital investment in energy saving projects are investigated in this chapter. In particular, the discounted cash flow techniques of *net present value and internal rate of return* are discussed in detail.

10.2 Fixed and Variable Costs

When appraising the potential costs involved in a project it is important to understand the difference between fixed and variable costs. Variable costs are those which vary directly with the output of a particular plant or production process, such as fuel costs. Fixed costs are those costs, which are not dependent on plant or process output, such as site-rent and insurance. The total cost of any project is therefore the sum of the fixed and variable costs. Example 10.1 illustrates how both fixed and variable costs combine to make the total operating cost.

Example 10.1

The capital cost of the DG set is Rs.9,00,000, the annual output is 219 MWh, and the maintenance cost is Rs.30,000 per annum. The cost of producing each unit of electricity is 3.50 Rs./kWh. The total cost of a diesel generator operating over a 5-year period, taking into consideration both fixed and variable cost is:

Item	Type of cost	Calculation	Cost
Capital cost of generator	Fixed	-	9,00,000
Annual maintenance	Fixed	30,000 x 5 (years)	1,50,000
Fuel cost	Variable	219,000 x 3.50 x 5	38,32,500
		Total cost	48,82,500

From Example 10.1, it can be seen that the fixed costs represent almost 21.5% of the total cost. In fact, the annual electricity output of 219 MWh assumes that the plant is operating with an average output of 50 kW. If this output were increased to an average of 70 kW, then the fuel cost would become Rs. 53,65,500 with the result that the fixed costs would drop to 16.37% of the total. Thus the average unit cost of production decreases as output increases.

The concept of fixed and variable costs can be used to determine the break-even point for a proposed project. The break-even point can be determined by using the following equation.

$$UC_{util} \times W_{av}, \times n = FC + (UC_{prod} \times W_{av} \times n)$$

Where,
UC_{util} is the unit cost per kWh of energy bought from utility (Rs./kWh)
UC_{prod} is the unit cost per kWh of produced energy (Rs./kWh)
FC is the fixed costs (Rs.)
W_{av} is the average power output (or consumption) (kW)
n is the number of hours of operation (hours).

Example 10.2

If the electricity bought from a utility company costs an average of Rs.4.5/kWh, the break-even point for the generator described in Example 1, when the average output is 50 kW is given by:

$$4.5 \times 50 \times n = (9,00,000 + 150000) + (3.5 \times 50 \times n)$$

$$n = 21000 \text{ hours}$$

If the average output is 70 kW, the break-even point is given by:

$$4.5 \times 70 \times n = (9,00.000 + 150000) + (3.50 \times 70 \times n)$$

$$n = 15000 \text{ hours}$$

Thus, increasing the average output of the generator significantly reduces the break-even time for the project. This is because the capital investment (i.e. the generator) is being better utilised.

10.3 Interest Charges

In order to finance projects, organizations often borrow money from banks or other leading organizations. Projects financed in this way cost more than similar projects financed from organisation's own funds, because interest charges must be paid on the loan. It is therefore important to understand how interest charges are calculated. Interest charges can be calculated by lending organization in two different ways: simple interest and compound interest.

(i) Simple interest: If simple interest is applied, then charges are calculated as a fixed percentage of the capital that is borrowed. A fixed interest percentage is applied to each year of the loan and repayments are calculated using the equation.

$$\text{Total Repayment Value (TRV)} = LV + (IR/100 + LV \times P)$$

Where TRV is the total repayment value (Rs.), LV is the value of initial loan (Rs.), IR is the interest rate (%), and P is the repayment period (years).

(ii) Compound interest: Compound interest is usually calculated annually (although this is not necessarily the case). The interest charged is calculated as a percentage of the outstanding loan at the end of each time period. It is termed 'compound' because the outstanding loan is the sum of the unpaid capital and the interest charges up to that point. The value of the total repayment can be calculated using the equation.

$$TRV = LV \times (1 + IR/100)^{P}$$

The techniques involved in calculating simple and compound interest are illustrated in Example 3 given below:

Example 10.3

A company borrows Rs.3,00,00,00 to finance a new boiler installation. If the interest rate is 10% per annum and the repayment period is 5 years, let us calculate the value of the total repayment and the monthly repayment value, assuming (i) simple interest and (ii) compound interest.

(i) Assuming simple interest:

Total repayment = 30,00,000 + (10/100 x 30,00,000 x 5) = Rs.45,00,000

Monthly repayment = 45,00,000/(5 x 12) = Rs.75,000

(ii) Assuming compound interest

Repayment at end of year 1 = 30,00,000 + (10/100 x 30,00,000) = Rs. 33,00,000

Repayment at end of year 2 = 33,00,000 + (10/100 x 33,00,000) = Rs. 36,30,000

Similarly, the repayments at the end of years 3, 4 and 5 can be calculated:

Repayment at end of year 3 = Rs. 39,93,000

Repayment at end of year 4 = Rs. 43,92,300

Repayment at end of year 5 = Rs. 48,31530

Alternatively, the following equation can be used to determine the compound interest repayment value.

$$\text{Total repayment value} = 30{,}00{,}000 \text{ x } (1+10/100)^5 = \text{Rs. } 48{,}31{,}530$$

$$\textit{Monthly repayment} = \frac{4831530}{5x12} = \textit{Rs.}80{,}525$$

It can be seen that by using compound interest, the lender recoups an additional Rs.33,1530. It is not surprisingly lenders usually charge compound interest on loans.

10.4 Simple Payback Period

This is the simplest technique that can be used to appraise a proposal. The simple payback period can be defined as 'the length of time required for the running total of net savings before depreciation to equal the capital cost of the project'. In theory, once the payback period has ended, all the project capital costs will have been recouped and any additional cost savings achieved can be seen as clear 'profit'. Obviously, the shorter the payback period, the more attractive the project becomes. The length of the maximum permissible payback period generally varies with the business culture concerned. In some companies, payback periods in excess of 3 years are considered acceptable.

The payback period can be calculated using the equation

$$\text{Simple payback period (years)} = \frac{\text{Capital cost of the project (in Rs.)}}{\text{Net Annual savings (in Rs.)}}$$

$$PB = \frac{CC}{AS}$$

Where PB is the payback period (years), CC is the capital cost of the project (Rs.), and AS is the annual net cost saving achieved (Rs.).

The annual net cost saving (AS) is the least savings achieved after all the operational costs have been met. Simple payback period is illustrated in Example 10.4.

Example 10.4

A new small cogeneration plant installation is expected to reduce a company's annual energy bill by Rs.4,86,000. If the capital cost of the new boiler installation is Rs.22,20,000 and the annual maintenance and operating costs are Rs. 42,000, the expected payback period for the project can be worked out as.

Solution

$$PB = 22,20,000/(4,86,000 - 42,000) = 5.0 \text{ years}$$

10. 5 Discounted Cash Flow Methods

The payback method is a simple technique, which can easily be used to provide a quick evaluation of a proposal. However, it has a number of major weaknesses:

- The payback method does not consider savings that are accrued after the payback period has finished.
- The payback method does not consider the fact that money, which is invested, should accrue interest as time passes. In simple terms there is a 'time value' component to cash flows. Thus Rs.1000 today is more valuable than Rs.1000 in 10 years' time.

In order to overcome these weaknesses a number of discounted cash flow techniques have been developed, which are based on the fact that money invested in a bank will accrue annual interest. The two most commonly used techniques are the 'net present value' and the 'internal rate of return' methods.

Net Present Value Method

The net present value method considers the fact that a cash saving (often referred to as a 'cash flow') of Rs.1000 in year 10 of a project will be worth less than a cash flow of Rs.1000 in year 2. The net present value method achieves this by quantifying the impact of time on any particular future cash flow. This is done by equating each future cash flow to its current value today, in other words determining the *present value* of any future cash flow. The *present value* (PV) is determined by using an assumed interest rate, usually referred to as a *discount rate.* Discounting is the opposite process to compounding. Compounding determines the future value of present cash flows, where" discounting determines the *present value* of future cash flows.

In order to understand the concept of *present value,* consider the case described in Example 10.4.

If instead of installing a new cogeneration system, the company invested Rs.22,20,000 in a bank at an annual interest rate of 8%, then:

The value of the sum at the end of year 1 = 22,20,000 + (0.08 x 22,20,000) = Rs.23,97,600

The value of the sum at the end of year 2 = 23,97,600 + (0.08 x 23,97,600) = Rs.25,89,408

The value of the investment would grow as compound interest is added, until after n years the value of the sum would be:

$$FV = D \times (1 + IR/100)^n$$

Where FV is the future value of investment in Rs., and D is the value of initial deposit (or investment) in Rs., IR is Interest Rate and n is number of years.

Example 10.5

The future value of the investment made at present, after 5 years will be:

$$FV = 22,20,000 \times (1 + 8/100)^5 = \text{Rs. } 32,61,908.40$$

So in 5 years the initial investment of 22,20,000 will accrue Rs.10,41,908.40 in interest and will be worth Rs.32,61,908.40. Alternatively, it could equally be said that Rs.32,61908.40 in 5 years time is worth Rs.22,20,000 now (assuming an annual interest rate of 8%). In other words the *present value* of Rs.32,61,908.40 in 5 years time is Rs.22,00,000 now.

The *present value* of an amount of money at any specified time in the future can be determined by the following equation.

$$PV = S \times (1 + IR/100)^{-n}$$

Where PV is the present value of S in n years time (Rs.), and S is the value of cash flow in n years time (Rs.).

The *net present value* method calculates the *present value* of all the yearly cash flows (i.e. capital costs and net savings) incurred or accrued throughout the life of a project, and summates them. Costs are represented as a negative value and savings as a positive value. The sum of all the present values is known as the *net present value* (NPV). The higher the *net present value, the* more attractive the proposed project.

The present value of a future cash flow can be determined using the equation above. However, it is common practice to use a *discount factor* (DF) when calculating present value. The discount factor is based on an assumed discount rate (i.e. interest rate) and can be determined by using equation.

$$DF = (1 + IR/100)^{-n}$$

The product of a particular cash flow and the discount factor is the present value.

$$PV = S \times DF$$

The values of various discount factors computed for a range of discount rates (i.e. interest rates) are shown in Table 10.1. The Example 10.6 illustrates the process involved in a net present value analysis.

Table 10.1 Computed Discount Factors								
Discount rate % (or interest rate %)								
Year	2	4	6	8	10	12	14	16
0	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
1	0.980	0.962	0.943	0.926	0.909	0.893	0.877	0.862
2	0.961	0.825	0.890	0.857	0.826	0.797	0.769	0.743
3	0.942	0.889	0.840	0.794	0.751	0.712	0.675	0.641
4	0.924	0.855	0.792	0.735	0.683	0.636	0.592	0.552
5	0.906	0.822	0.747	0.681	0.621	0.567	0.519	0.476
6	0.888	0.790	0.705	0.630	0.564	0.507	0.456	0.410
7	0.871	0.760	0.665	0.583	0.513	0.452	0.400	0.354
8	0.853	0.731	0.627	0.540	0.467	0.404	0.351	0.305
9	0.837	0.703	0.592	0.500	0.424	0.361	0.308	0.263
10	0.820	0.676	0.558	0.463	0.386	0.322	0.270	0.227
11	0.804	0.650	0.527	0.429	0.350	0.287	0.237	0.195
12	0.788	0.625	0.497	0.397	0.319	0.257	0.208	0.168
13	0.773	0.601	0.469	0.368	0.290	0.229	0.182	0.145
14	0.758	0.577	0.442	0.340	0.263	0.205	0.160	0.125
15	0.743	0.555	0.417	0.315	0.239	0.183	0.140	0.108
16	0.728	0.534	0.394	0.292	0.218	0.163	0.123	0.093
17	0.714	0.513	0.371	0.270	0.198	0.146	0.108	0.080
18	0.700	0.494	0.350	0.250	0.180	0.130	0.095	0.069
19	0.686	0.475	0.331	0.232	0.164	0.116	0.083	0.060
20	0.673	0.456	0.312	0.215	0.149	0.104	0.073	0.051

Example 10.6

Using the net present value analysis technique, let us evaluate the financial merits of the proposed projects shown in the Table below. Assume an annual discount rate of 8% for each project.

	Project – 1	**Project - 2**
Capital cost (Rs.)	30 000.00	30 000.00
Year	Net annual saving (Rs.)	Net annual saving (Rs.)
1	+6 000.00	+6 600.00
2	+6 000.00	+6 600.00
3	+6 000.00	+6 300.00
4	+6 000.00	+6 300.00
5	+6 000.00	+6 000.00
6	+6 000.00	+6 000.00
7	+6 000.00	+5 700.00
8	+6 000.00	+5 700.00
9	+6 000.00	+5 400.00
10	+6 000.00	+5 400.00
Total net saving at end of year 10	+ 60 000.00	+ 60 000.00

Solution

The annual cash flows should be multiplied by the annual discount factors for a rate of 8% to determine the annual *present values*, as shown in the Table below:

Year	Discount Factor for 8% (a)	Project 1		Project 2	
		Net savings (Rs.) (b)	Present value (Rs.) (a x b)	Net savings (Rs.) (c)	Present value (Rs.) (a x c)
0	1.000	-30 000.00	-30 000.00	-30 000.00	-30 000.00
1	0.926	+6 000.00	+5 556.00	+6 600.00	+6 111.60
2	0.857	+6 000.00	+5 142.00	+6 600.00	+5 656.20
3	0.794	+6 000.00	+4 764.00	+6 300.00	+5 002.20
4	0.735	+6 000.00	+4 410.00	+6 300.00	+4 630.50
5	0.681	+6 000.00	+4 086.00	+6 000.00	+4 086.00
6	0.630	+6 000.00	+3 780.00	+6 000.00	+3 780.00
7	0.583	+6 000.00	+3 498.00	+5 700.00	+3323.10
8	0.540	+6 000.00	+3 240.00	+5 700.00	+3 078.00
9	0.500	+6 000.00	+3 000.00	+5 400.00	+2 700.00
10	0.463	+6 000.00	+2 778.00	+5 400.00	+2 500.20
		NPV = +10 254.00		**NPV = +10 867.80**	

It can be seen that over a 10 year life-span the *net present value* for Project 1 is Rs.10,254.00, while for Project 2 it is Rs.10,867.80. Therefore Project 2 is the preferential proposal.

The whole credibility of the net present value method depends on a realistic prediction of future interest rates, which can often be unpredictable. It is prudent therefore to set the discount rate slightly above the interest rate at which the capital for the project is borrowed. This will ensure that the overall analysis is slightly pessimistic, thus acting against the inherent uncertain ties in predicting future savings.

Internal rate of return method

It can be seen from Example 10.6 that both projects returned a positive net present value over 10 years, at a discount rate of 8%. However, if the discount rate were reduced there would come a point when the net present value would become zero. It is clear that the discount rate which must be applied, in order to achieve a net present value of zero, will be higher for Project 2 than for Project 1. This means that the average rate of return for Project 2 is higher than for Project 1, with the result that Project 2 is the better proposition.

The discount rate which achieves a net present value of zero is known as the internal rate of return (IRR). The higher the internal rate of return, the more attractive the project.

Example 10.7 illustrates how an *internal rate of return* analysis is performed.

Example 10.7

A proposed project requires an initial capital investment of Rs.20 000. The cash flows generated by the project are shown in the table below:

Year	Cash flow (Rs.)
0	-20,000.00
1	+6000.00
2	+5500.00
3	+5000.00
4	+4500.00
5	+4000.00
6	+4000.00

Given the above cash flow data, let us find out the internal rate of return for the project.

Solution

	Cash flow (Rs.)	8% discount rate		12% discount rate		16% discount rate	
		Discount factor	Present value (Rs.)	Discount factor	Present value (Rs.)	Discount factor	Present value (Rs.)
0	-20000	1.000	-20000	1.000	-20000	1.000	-20000
1	6000	0.926	5556	0.893	5358	0.862	5172
2	5500	0.857	4713.5	0.797	4383.5	0.743	4086.5
3	5000	0.794	3970	0.712	3560	0.641	3205
4	4500	0.735	3307.5	0.636	3862	0.552	2484
5	4000	0.681	2724	0.567	2268	0.476	1904
6	4000	0.630	2520	0.507	2028	0.410	1640
		NPV = 2791		NPV = 459.5		NPV = -1508.5	

It can clearly be seen that the discount rate which results in the net present value being zero lies somewhere between 12% and 16%.

For12% discount rate, NPV is positive; for 16% discount rate, NPV is negative. Thus for some discount rate between 12 and 16 percent, present value benefits are equated to present value costs. To find the value exactly, one can interpolate between the two rates as follows:

$$\text{Internal rate of return} = 0.12 + (0.16\text{-}0.12) \times \frac{459.5}{(459.5\text{-}(\text{-}1508.5))}$$

$$\text{Internal rate of return} = 0.12 + (0.16\text{-}0.12) \times \frac{459.5}{(459.5+1508.5)} = 12.93\%$$

Thus the internal rate of return for the project is 12.93 %. At first sight both the net present value and internal rate of return methods look very similar, and in some respects are. Yet there is an important difference between the two. The net present value method is essentially a comparison tool, which enables a number of projects to be compared, while the internal rate of return method is designed to assess whether or not a single project will achieve a target rate of return.

Profitability index

Another technique, which can be used to evaluate the financial viability of projects, is the profitability index. The profitability index can be defined as:

$$\text{Profitability index} = \frac{\text{Sum of the discounted net savings}}{\text{Capital costs}}$$

The higher the profitability index, the more attractive the project.

The application of profitability index is illustrated in Example 10.8.

Example 10.8

Determine the profitability index for the projects outlined in Example 10.6

$$\text{For Project 1: Profitability index} = \frac{10254}{30000} = 0.342$$

$$\text{For Project 2: Profitability index} = \frac{10867}{30000} = 0.362$$

Project 2 is therefore a better proposal than Project 1.

10.6 Factors Affecting Analysis

Although the Examples 10.5 and 10.6 illustrate the basic principles associated with the financial analysis of projects, they do not allow for the following important considerations:

- The capital value of plant and equipment generally depreciates over time
- General inflation reduces the value of savings as time progresses. For example, Rs.1000 saved in 1 year's time will be worth more than Rs.1000 saved in 10 years time.

The capital depreciation of an item of equipment can be considered in terms of its salvage value at the end of the analysis period. The Example 10.9 illustrates the point.

Example 10.9

It is proposed to install a heat recovery equipment in a factory. The capital cost of installing the equipment is Rs.20,000 and after 5 years its salvage value is Rs.1500. If the savings accrued by the heat recovery device are as shown below, we have to find out the net present value after 5 years. Discount rate is assumed to be 8%.

Data

Year	1	2	3	4	5
	7000	6000	6000	5000	5000

Solution

Year	Discount Factor for 8% (a)	Capital Investment (Rs.) (b)	Net Savings (Rs.) (c)	Present Value (Rs.) (a) x (b + c)
0	1,000	-20,000.00		-20,000.00
1	0.926		+7000.00	+6482.00
2	0.857		+6000.00	+5142.00
3	0.794		+6000.00	+4764.00
4	0.735		+5000.00	+3675.00
5	0.681	+1,500.00	+5000.00	+4426.50
				NPV = +4489.50

It is evident that over a 5-year life span the net present value of the project is Rs.4489.50. Had the salvage value of the equipment not been considered, the net present value of the project would have been only Rs.3468.00.

Real value

Inflation can be defined as the rate of increase in the average price of goods and services. In some countries, inflation is expressed in terms of the retail price index (RPI), which is determined centrally and reflects average inflation over a range of commodities. Because of inflation, the real value of cash flow decreases with time. The real value of sum of money (S) realised in n years time can be determined using the equation.

$$RV = S \times (1 + R/100)^{-n}$$

Where RV is the real value of S realized in n years time. S is the value of cash flow in n years time and R is the inflation rate (%).

As with the discount factor it is common practice to use an inflation factor when assessing the impact of inflation on a project. The inflation factor can be determined using the equation.

$$IF = (1 + R/100)^{-n}$$

The product of a particular cash flow and inflation factor is the real value of the cash flow.

$$RV = S \times IF$$

The application of inflation factors is considered in Example 10.10.

Example 10.10

Recalculate the net present value of the energy recovery scheme in Example 10.9, assuming the discount rate remains at 8% and that the rate of inflation is 5%.

Solution

Because of inflation; Real interest rate = Discount rate – Rate of inflation

Therefore Real interest rate = 8-5 = 3%

Year	Capital Investment (Rs.)	Net real Savings (Rs.)	Inflation Factor For 5%	Net real Savings (Rs.)	Real Discount Factor For 3%	Present Value (Rs.)
0	-20,000.00		1.000	-20,000.00	1.000	-20,000.00
1		+7000.00	0.952	+6664.00	0.971	+6470.74
2		+6000.00	0.907	+5442.00	0.943	+5131.81
3		+6000.00	0.864	+5184.00	0.915	+4743.36
4		+5000.00	0.823	+4145.00	0.888	+3654.12
5	+1500.00	+5000.00	0.784	+5096.00	0.863	+4397.85
						NPV = +4397.88

The Example 10.10 shows that when inflation is assumed to be 5%, the net present value of the project reduces from Rs.4489.50 to Rs.4397.88. This is to be expected, because general inflation will always erode the value of future 'profits' accrued by a project.

Solved Example:

(i) Two energy conservation projects have been proposed.

For the first project, a capital investment of Rs.1,50,000/- is required and the net annual saving is Rs. 50,000/- for 5 years. The salvage value at the end of 5 years for the first project is Nil.

For the second project, a capital investment of Rs. 1,50,000/- yields savings of Rs. 50,000/- for first 2 years each and Rs. 60,000/- for next 3 years each. The salvage value at the end of 5 years for the second project is Rs. 10,000/-. Determine:

a) Net present value for both the projects with a discount factor of 9%.
b) Profitability index for both the projects with a discount factor of 9%.
c) Internal rate of return for both the projects.

(ii) For a VFD retrofit in a compressed air system with an initial investment of Rs.2.55 lakhs the annual savings are Rs.58,000/-

The NPV of the project over a six year period for 8% discount rate is Rs.13,134.

The NPV of the project over a six year period for 10% discount rate is Rs.(-) 2468

Calculate the exact discount rate for NPV to be zero.

Solution:

(i)

a) NPV for project 1 = – 150000 + 50000/1.09 + 50000/$(1.09)^2$ + 50000/$(1.09)^3$ + 50000/$(1.09)^4$ + 50000/$(1.09)^5$ = + 44483

NPV for project 2 = – 150000 + 50000/1.09 + 50000/$(1.09)^2$ + 60000/$(1.09)^3$ + 60000/$(1.09)^4$ + 60000/$(1.09)^5$ + 10000/$(1.09)^5$ = + 72287

b) Profitability index for project 1 = 44483 / 150000 = 0.297

Profitability index for project 2 = 72287 / 150000 = 0.481

c) IRR for project 1 = 19.86 %

IRR for project 2 = 25.07 %

(ii) The positive and negative NPV's which are closest to zero discount rates are Rs (+)13,134 corresponding to 8% discount rate and Rs (-) 2,468 corresponding to10% discount rate.

Hence the exact discount rate = (0.08+(0.1-0.08)*(13134)/(13134-(-468))*100

= 9.68 %

QUESTIONS	
S-1	Define ROI.
S-2	Explain why a project with a high IRR is not necessarily more attractive than a project with a lower IRR.
S-3	You are investing Rs.100 in a bank. The bank gives 10% interest per year for two years. What is the present value and what is the future value?
S-4	What do you understand by the term "present value of money"?
S-5	What are the advantages of simple pay back method?
S-6	What is the main draw back of simple pay back method?
S-7	What is profitability index? How is it calculated?
S-8	The cost of a heat exchanger is Rs.1.00 lakh. Calculate the simple pay back period considering an annual saving potential of Rs.60,000/- and annual operating cost of Rs.15,000/-.
S-9	Calculate simple pay back period for a boiler that cost Rs.75.00 lakhs to purchase and Rs.5 lakhs per year on an average to operate and maintain and is expected to save Rs.30 lakhs.
S-10	What is the objective of carrying out sensitivity analysis?
L-1	What are the factors affecting financial analysis while computing NPV and IRR?
L-2	It is proposed to install a heat recovery equipment in a food processing industry. The capital cost of the equipment is Rs 20,000/. The savings accrued by the unit are constant and Rs 5,000/- annually. The discount rate is 8%. Calculate the Net Present Value (NPV) for 5 years.
N-1	A waste heat recovery system can be installed in a furnace, which will cost Rs. 6,50,000/- to install. This system is expected to have a useful life of 6 years. The salvage value will be Rs. 1,00,000/-. The system will reduce the energy cost by Rs. 2,00,000/- per year, when it operates at full capacity. However the plant will be operating at partial capacity for the first 3 years and the annual energy savings will be at 70% of the energy cost savings at full capacity (Rs. 1,40,000/-). The new system will entail a maintenance cost of Rs. 10,000/- per year for the first 3 years and Rs.12,000 per year for the next 3 years. A major overhaul is required in the 3rd year, which will cost Rs 1,00,000/- 1. If discount rate is 9%, calculate the NPV and find out whether this energy conservation measure is profitable. 2. What is the profitability index for the project?

REFERENCES

1. Energy Management, Supply and Conservation, Dr. Clive Beggs, .Butterworth Heinemann

Chapter 11

Performance Assessment of Thermal Power Station

Learning Objectives

In this chapter you will learn about

- Performance terms and definitions
- Theory of Rankine cycle
- The purpose of re-heat and regenerative systems
- Major areas/equipment in Thermal Power Station
- Performance assessment of Coal handling plant, Mills, Boilers, Heaters, Turbines, Draft system, pumps and condensers .

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Estimate the station gross heat rate, net heat rate and specific fuel consumption**
- ✓ **Perform heat balance of Milling system**
- ✓ **Assess the performance of CHP, Mills, Draft system, Heaters and Condensers etc.**
- ✓ **Estimate the turbine stage efficiency**

11. ENERGY PERFORMANCE ASSESSMENT OF THERMAL POWER STATION

11.1 Introduction

Coal fired thermal power plants generate major portion of India's electricity. In a typical thermal power plant the raw coal is crushed and pulverized in a mill to a size finer than face powder. The primary air supply dries and transports the coal into the boiler furnace. The coal burns in the furnace to generate superheated steam which drives a turbine connected to an alternator to generate electricity.

Some Useful Definitions in Indian Context

Auxiliary Energy Consumption in relation to a period means the quantum of energy consumed by auxiliary equipment of the generating station and transformer losses within the generating station, and shall be expressed as a percentage of the sum of gross energy generated at the generator terminals of all the units of the generating station;

Financial Year – Financial year means the year beginning on the 1st day of April and ending on the 31st March following;

Gross Heat Rate or 'GHR' means the heat energy input in kcal required to generate one kWh of electrical energy at generator terminals;

Net Heat Rate or NHR – The heat energy in kcal, input to a Generating Station to deliver one kWh at the switchyard.

$$\textbf{Plant Load Factor or 'PLF'} = \frac{\text{Energy generated in a given period (MWh)}}{\text{Total capacity (MW) x total hours in the same period}} \times 100$$

The components of a typical power plant are depicted in the Figure 11.1.

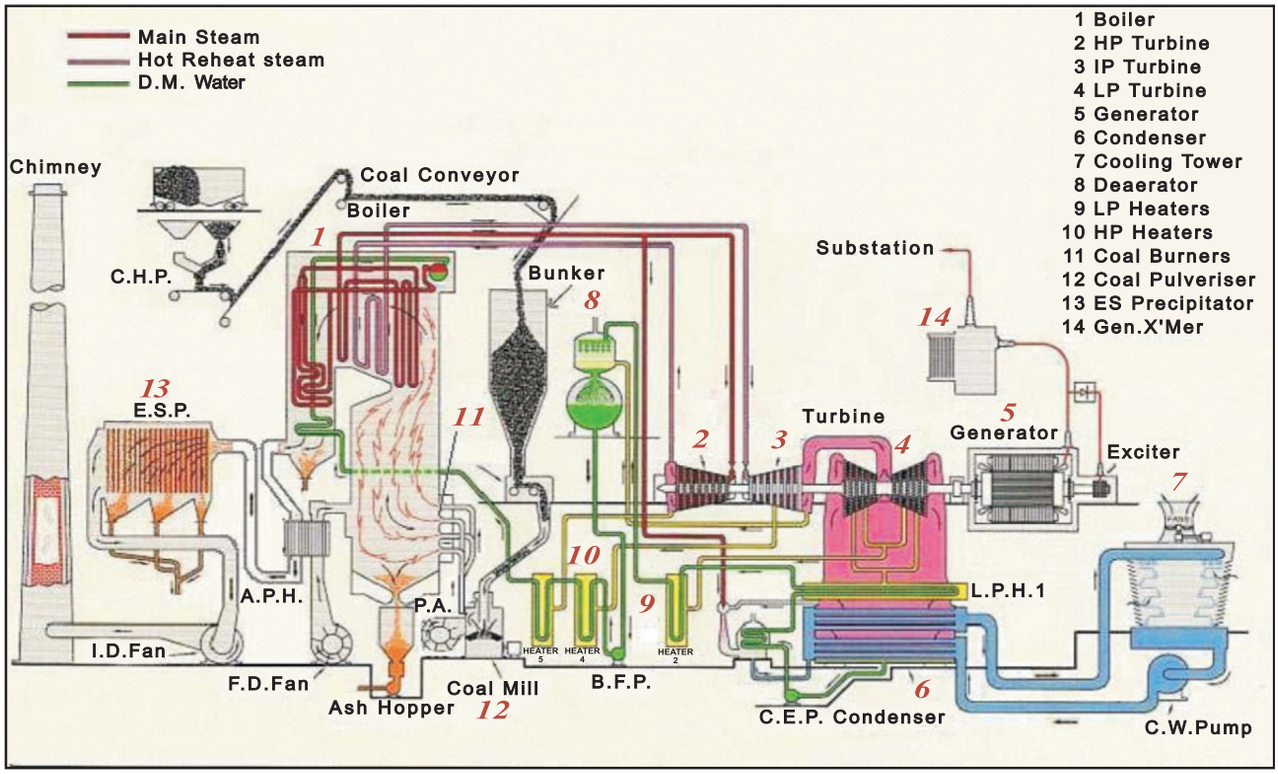


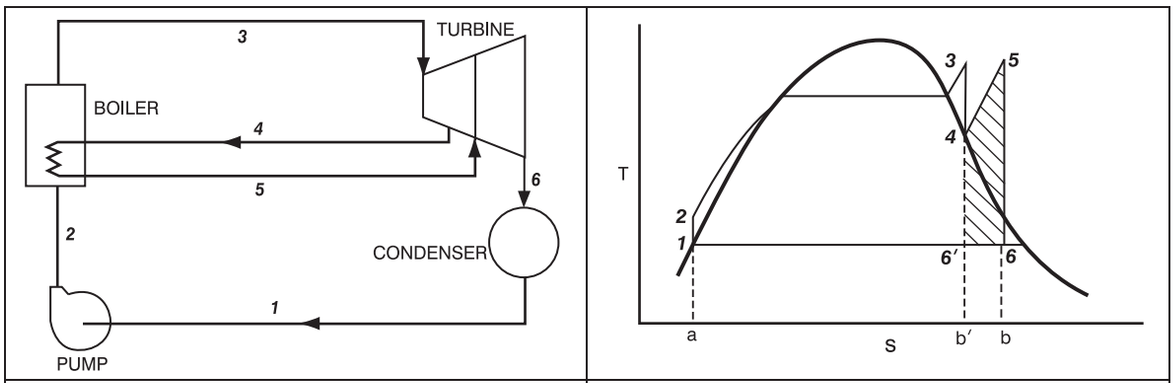
Figure 11.1 Components of a Typical Power Plant

The power plants operate on a modified Rankine cycle with reheat and regenerative systems.

Reheat

Reheating the steam after it has partially expanded through the turbine increases the average temperature of heat addition and cycle efficiency. For a typical 210 MW power plant, (Figure 11.4) major part of the superheated steam from HP turbine exhaust at 36 kg/cm^2, at around 336 ^{o}C is brought to the reheat section of the boiler, where the temperature is raised to approximately the main steam superheat temperature of 535 ^{o}C. The hot reheat steam is then returned to the intermediate pressure turbine to complete its expansion. This helps in extracting greater amount of work from the steam.

Figure 11.2 shows the equipment arrangement and the *T-S* diagram for the single reheat cycle. The additional work as a result of reheating is shown by the shaded area bounded by points 4-5-6-6'. The additional unavailable heat that must be rejected is shown by the area bounded by b ' - 6 ' - 6 - b. The gain in net work, therefore efficiency, is realized as the heat added, is greater than the additional heat rejected. Among other advantages, this prevents the vapor from condensing during its expansion which can seriously damage the turbine blades.



Reheat	Effect of reheating on TS diagram

Figure 11.2 Reheat Cycle

Regenerative Feed Water Heating

In a typical power plant cycle shown in Figure 11.3 the regenerative portion utilizes partially expanded steam extracted from the turbine at various points to heat the condensate and feed water through HP/ LP heaters on its way back to the boiler or steam generator. A part of the steam (Figure 11.4) is drawn at 36 kg/cm^2 from the high pressure (HP) turbine and at 16 kg/cm^2 from the intermediate pressure (IP) turbine for the HP heaters. For the LP heaters the steam is extracted from various stages of low pressure (LP) turbine. In large power plants as many as eight stages may be employed. What this means is that incremental steps of heat transfer to the feed water increases the cycle efficiency as against having the entire heat transfer taking place within the boiler.

Increasing the average temperature of heat addition can also be accomplished by increasing the temperature of the feed water entering the boiler. To realize a gain in efficiency, heat from within the cycle is used to elevate the feed water temperature. This can be done by extracting a portion of the partially expanded steam from the turbine and directing it to a heat exchanger that heats the feed water to the boiler. This process is called regenerative feed water heating.

11.2 Purpose of the Performance Test

The purpose of the performance test is to arrive at the efficiency parameters for the power plant as a whole and also the various components of the power plant.

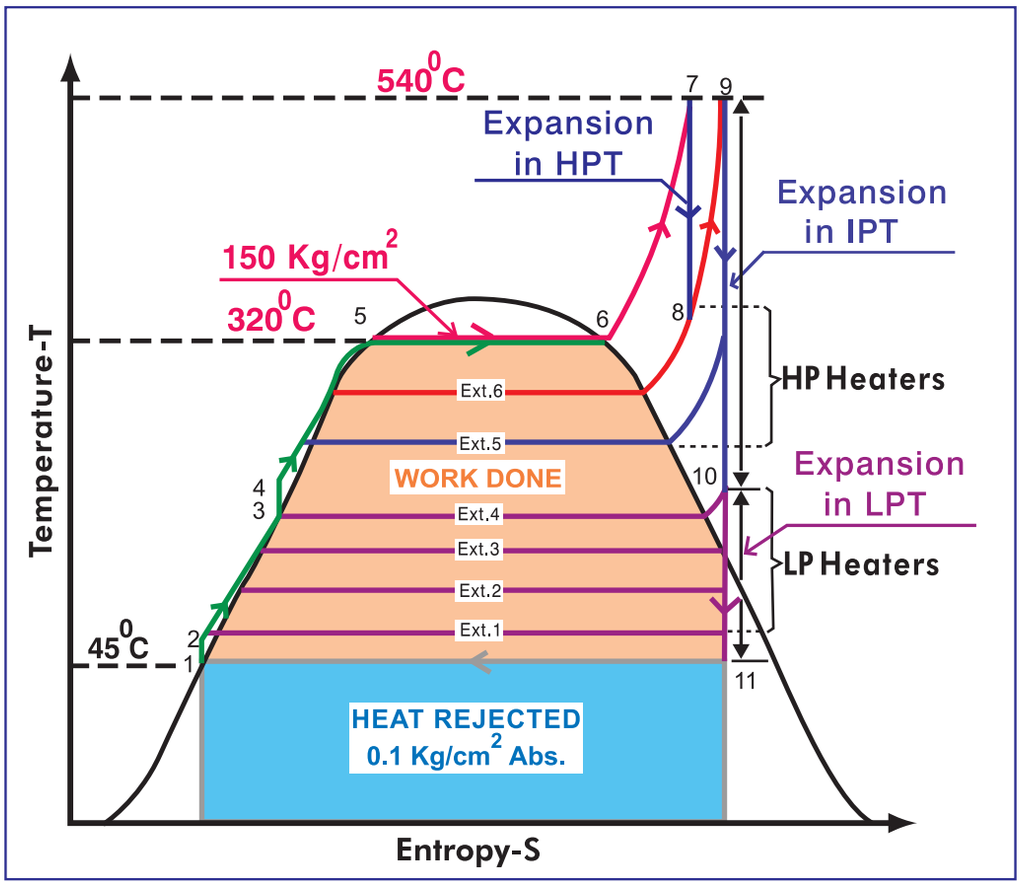


Figure 11.3 Regenerative Feedwater Heating

11.3 Performance Terms and Definitions

This section gives a brief detail of various performance parameters dealing with the power plants.

Gross heat rate (Q_e), kcal/kWh

$$= \frac{\text{Fuel consumption, kg} \times \text{GCV of fuel, kcal/kg}}{\text{Generator output, kWh}}$$

Net heat rate refers to the heat rate after deducting the auxiliary power consumption

$$\text{Net heat rate } (Q_e), \text{kcal/kWh} = \frac{\text{Gross heat rate, kcal / kWh}}{1 - (\%\ \text{auxiliary power consumption} / 100)}$$

Overall efficiency

The term "overall efficiency" as defined solely for power generation only considers the useful electrical output to effective heat input to the power plant. Therefore, the equation for efficiency is

$$\text{Overall efficiency, } \% = \frac{\text{Generator output, kW}}{\text{Mass flow rate of fuel, kg/s} \times \text{Gross calorific value, kJ/kg}} \times 100$$

The overall efficiency of power plant can also be determined from Q_e as follows

$$\text{Overall efficiency, \%} = \frac{860}{\text{Gross heat rate, kcal/kWh}} \times 100$$

Specific fuel consumption

$$\text{Specific fuel consumption, kg/kWh} = \frac{\text{Total fuel consumption, kg}}{\text{Gross generation, kWh}}$$

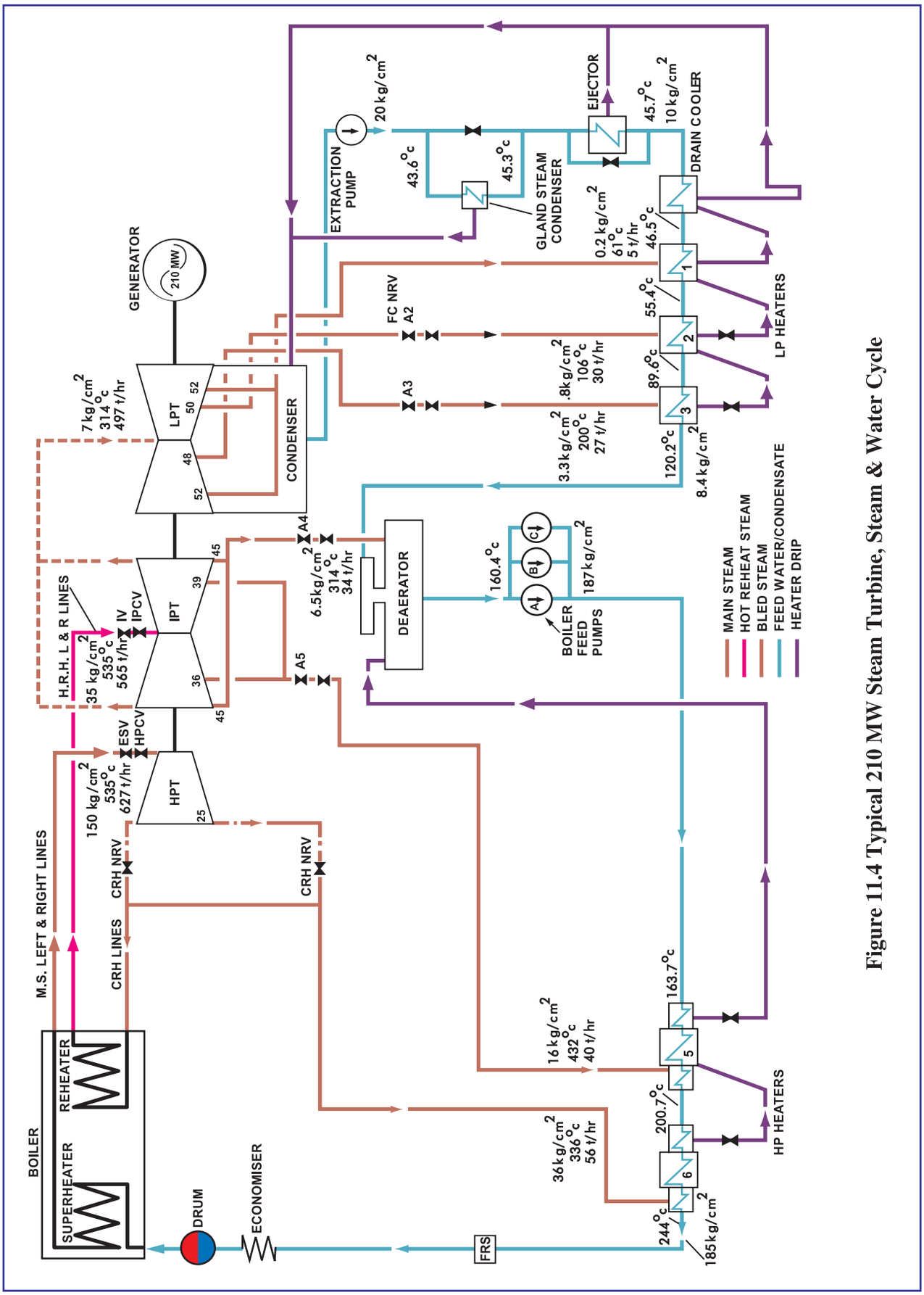


Figure 11.4 Typical 210 MW Steam Turbine, Steam & Water Cycle

11.4 Major Area/Equipment in Thermal Power Plant

The major areas for conducting performance test in thermal power plants are:

- Fuel handling system and preparation (E.g.: Coal handling system and coal mills)
- Boilers and its associated parts
- Turbines and its associated parts
- Draft system /Fans (ID fans, FD fans, PA fans and other fans.)
- Condensers
- Water pumping systems (Boiler feed water pumping system, Condensate extraction pumping system, DM water pumping system, Make up water pumping, Raw water pumping system, etc).

Table 11.1. Typical auxiliary power consumption in power plant

Equipment Ref.	500 MW		210 MW		110 MW	
	% Gen	% APC	% Gen	% APC	% Gen	% APC
BFP	0.00*	0.00*	2.70	33.60	2.94	24.50
CEP	0.40	5.70	0.27	3.34	0.36	3.00
CWP	1.00	14.20	0.66	8.31	1.26	10.50
IDF	1.30	18.70	1.26	15.80	1.71	14.23
PAF	0.60	8.50	0.68	6.50	1.78	14.46
FDF	0.30	4.10	0.40	5.00	0.26	2.13
Mills	0.60	8.20	0.58	7.23	0.83	6.92
CT fans	0.23	3.20	0.32	3.54	0.48	4.00
Air Comp.	0.08	1.20	0.12	1.56	0.24	2.00
A/C Plant	0.04	0.50	0.08	0.94	0.11	0.92
CHP	0.12	1.70	0.14	1.70	0.29	2.41
AHP	0.09	1.20	0.13	1.66	0.31	2.54
Lighting	0.06	0.80	0.08	1.00	0.08	0.68
others	2.23	31.90	0.60	7.44	1.36	11.32
APC	**7.00**	**100.00**	**8.00**	**100.00**	**12.00**	**100.00**

* turbine driven

BFP – Boiler feed pump, CEP – Condensate extraction pump, CWP - Cooling water pump, IDF – Induced draft fan, PAF – Primary air fan, FDF – Forced draft fan, CT fans – Cooling tower fans, CHP – Coal handling plant, AHP – Ash handling plant, APC – Auxiliary power consumption and % Gen - % Generation.

Table 11.2. CERC norms (2014-19) for Auxiliary Energy Consumption of coal-based generating stations

S.No.	Capacity	With Natural Draft cooling tower or with-out cooling tower	With Induced Draft cooling towers	With Direct cooling; Air cooled con-densers with mechanical draft fans	Indirect cooling system employing jet condensers with pressure recovery turbine & natural draft tower
1.	200 MW series	8.5%	9.0%	9.5%	9.0%
2.	300/330/350/500 MW and above having				
(i)	steam driven boiler feed pumps	5.25%	5.75%	6.25%	5.75%
(ii)	electrically driven boiler feed pumps	7.75%	8.25%	8.75%	8.25%

11.5 Coal Handling Plant

Coal handling plant is one of the important energy consumers in thermal power plants and contains the following energy consuming equipments.

- Crushers
- Conveyors
- Feeders
- Tipplers

The major objectives of coal handling plant energy audit are

- To evaluate specific energy consumption of the CHP equipments (kWh/ton of coal)
- To evaluate percentage power consumption of CHP with respect to total auxiliary power consumption
- To analyse the crushed coal size and rejects

Performance measurements

- Carry out power measurements of conveyors, crushers and feeders etc. In the absence of energy meters, take readings from on-line panel instruments for current, voltage, power factor etc., portable instrument can also be used for power measurements.
- The coal flow rate can be recorded using weigh feeders.

Table 11.3. Typical specific energy consumption of the CHP

Equipment	Rated capacity		Measured		SEC
	Motor kW	Coal TPH	Motor kW	Coal pro-cessed TPH	kWh/ Ton
Conveyor	280	1400	144	1134	0.127
Crusher	597	875	253	567	0.446

Conveyor	500	1400	172.4	1362	0.126
Grizzly Feeder	45	875	16.9	620	0.027
Paddle Feeder	37	875	17.2	620	0.027
Etc.					
Total					**1.2**

Table 11.4. Auxilliary power consumption of CHP

Parameter	Unit	Value
Total auxiliary power consumption of the station in a period	MU	
Total power consumption by the CHP plant in the same period	MU	
% Power consumption of CHP with respect to total auxiliary power consumption	%	

Performance of Crushers

- Observe and compare the operation of crushers and their throughput, hours of operation, specific power consumption, etc
- Carryout the size analysis and compare with the design or optimum values
- If significant proportion of coal >20 mm size is observed on the downstream of crusher, it may lead to substantial increase in power consumption of coal mills.

11.6 Coal Mills

The major objectives of coal mill energy audit are

- To evaluate specific energy consumption of the mills. (kWh/ton of coal)
- To establish air to coal ratio of the mills (ton of air per ton of coal)
- To perform heat balance of the mills
- To analyse the coal fineness and mill rejects

Performance measurements

- Carry out power measurements of mills, PA fans, seal air fans, etc,. In the absence of energy meters, take readings from on-line panel instruments for current, voltage, power factor etc. For LT equipment, portable instrument can be used for power measurements.
- Coal flow to be established by dirty Pitot tube test (to be carried out on Pulverised coal lines). This also helps to identify unbalancing/choking is occurring in flow in the Pulverised Coal lines. The on line coal flow values if available, may also to be taken by appropriate coal feeder calibration.
- Airflow to be established per PA fan by clean air Pitot tube method

Air to coal ratio of the mill

$$Air\ to\ coal\ ratio\ of\ the\ mill = \frac{Air\ \ flow\ rate, kg/hr}{Coal\ \ flow\ rate, kg/hr}$$

Specific energy consumption of the mill

$$Specific\ \ energy\ consumption\ of\ the\ mill = \frac{Electricity\ consumption, kW}{Coal\ \ flow\ rate, TPH}$$

Example 11.1. Determine the air fuel ratio and specific energy consumption of mill from the following data

Coal flow rate = 48.64 TPH
Air flow rate = 112.6 TPH
Energy consumption = 351 kW

$$Air\ to\ coal\ ratio\ \ of\ the\ mill = \frac{Air\ \ flow\ rate, kg/hr}{Coal\ \ flow\ rate, kg/hr}$$

= 112.6 / 48.64

= 2.31 kg of air/kg of coal

$$Specific\ \ energy\ consumption\ the\ mill, kWh/T = \frac{Electricy\ consumption, kW}{Coal\ \ flow\ rate, TPH}$$

= 351 / 48.64
= 7.22 kWh/T of coal

Mill Heat Balance

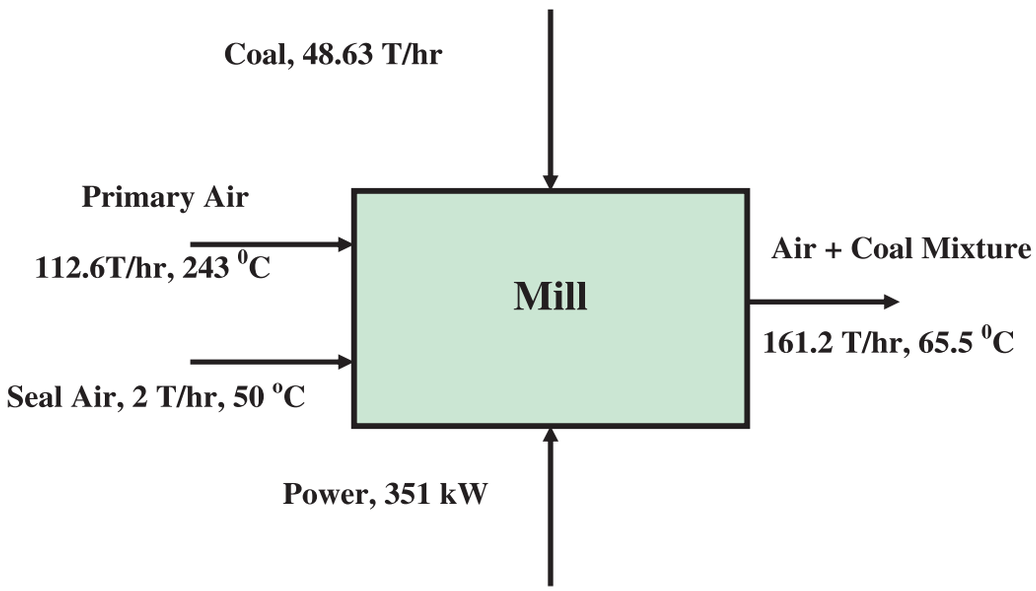


Figure 11.5 Schematic of mill heat balance

Primary air – used for drying and conveying the coal, Seal air – pressurised air used to seal the motor shaft of coal mill to avoid leakage.

Example 11.2. Coal Mill Heat Balance

Total PA flow	= 112.6 TPH
Inlet PA temperature	= 243 ^{o}C
Seal air flow	= 2.0 TPH
Seal air temperature	= 50 ^{o}C
Coal flow rate	= 48.63 TPH
Power consumption	= 351 kW
Air and coal mix flow	= 161.23 TPH
Air and coal mix temperature	= 65.5 ^{o}C
Ambient temperature	= 35 ^{o}C
Reference temperature	= 0 ^{o}C

Heat input

Sensible heat of primary air I/L = Flow x specific heat x (T - t)
= 112600 x 0.24 x (243 – 0)
= 6566832 kcal/hr

Sensible heat of seal air = Flow x specific heat x (T - t)
= 2000 x 0.24 x (50 – 0)
= 24000 kcal/hr

Heat equivalent of power = Power consumption x 860
= 351 x 860
= 301860 kcal/hr

Total heat input = 6566832 + 24000 + 301860
= 6892692 kcal/hr

Heat output

Sensible heat of coal air mix = Flow x specific heat x (T - t)
= 161230 x 0.24 x (65.5 – 0)
= 2534536 kcal/hr

Heat carried away by moisture (by difference) = Total heat input – Heat of coal air mix.
= 6892692 – 2534536
= 4358156 kcal/hr

Total moisture removed = 4358156 / 540
= 8071 kg/hr

% Moisture removed = 8071 / 48630
= 16.59 %

Analysis of mill fineness and mill rejects

For a typical PF coal fired boiler, over 80% of fine coal should pass through 200 mesh for good combustion efficiency. Mill performance is monitored by fineness of mill output at regular intervals and when there is a fall, maintenance practices are reviewed.

Mill rejects

Mill rejects are an indication of mill performance. It is desirable to carryout the detailed analysis of mill rejects (table 11.5), compare with the other mills and coal quality. The variation in mill rejects (among the mills) could be due to variation in performance of mill due to mill internal status, fuel handling, etc.

Table 11.5. Mill rejects reporting

Mill No	Fuel input, TPH	Fuel Reject, TPH	Mill reject %	Calorific value of the reject	Remarks
Mill No #					

11.7 Boiler

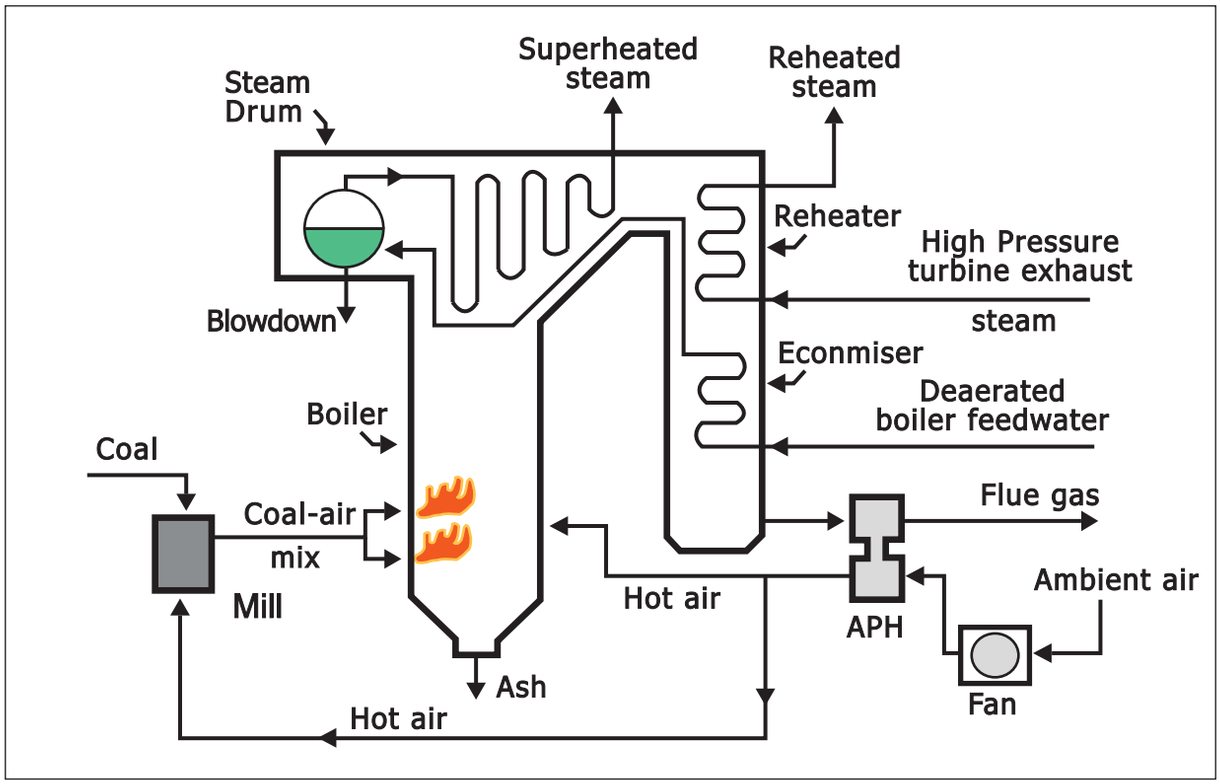


Figure 11.6 Schematic diagram of a utility boiler

The brief specifications of a typical boiler used in 210 MW plant is given in the table 11.6.

Table 11.6. Typical boiler specifications for a 210 MW unit

Particulars	Unit	Details at Normal Continuous Rating (NCR)
Type		Water tube single drum
Capacity	TPH	627.32
Main Steam pressure	kg/cm^2	155
Main Steam temperature	^0C	540
Boiler efficiency	%	87.16
Super heater outlet flow	TPH	627.32
Reheater outlet flow	TPH	565.6
GCV of coal	kcal/kg	4350
Coal consumption	TPH	106.2
Total combustion air	TPH	822
Reheater outlet temperature	^0C	540
Water-economizer inlet temperature	^0C	241
Water-economizer outlet temperature	^0C	280
Oxygen content at economizer outlet	%	4.23

For performance evaluation of boiler, refer to Chapter – 1 of this book.

Performance evaluation of economizer and air preheater

Example 11.3. Performance of Economiser and Air Pre-Heater

Heat Exchanger Effectiveness Operating Parameters	UNIT	Design
Generation	MW	210
Primary Air flow through APH	TPH	110
FW flow to economiser	TPH	609
Gas temp at economiser in	^oC	473
Gas temp at economiser out	^oC	379
F W temp at economiser in	^oC	244
F W temp at economiser out	^oC	274
Gas temp at APH out	^oC	146
Gas temp at APH in	^oC	379
Primary air temp at APH in	^oC	35
Primary air temp at APH out	^oC	354

Economiser Effectiveness

= (F. W temp out – FW temp in) / (F. Gas temp ECO in – F. Water temp in)
= (274 – 244) / (473 – 244)
= 13.1 %

Heat Pick-up in Economiser :

= Feed water Flow x 1 x (FW temp. out – FW temp. in)
= 609000 x 1 x (274 – 244)
= 18270 MCal/hr

APH Effectiveness

= (Air temp APH out – Air Temp APH in)/(Flue Gas temp. APH in – Air temp. APH in)
= (354 – 35) / (379 – 35)
= 92.7 %

Heat pick-up in APH:

= Air Flow x 0.24 x (Air temp APH out – Air temp. APH in)
= 110000 x 0.24 x (354 – 35)
= 8422 MCal/hr

Air leakage estimation:

The following gives the air leakage in to the air preheaters (APH) system if the Oxygen % is measured at the entry and exit of the air preheaters (APH)

$$APH\ leakage\ \% = \frac{(O_2\%\ in\ the\ gas\ leaving\ the\ APH - O_2\%\ in\ the\ gas\ entering\ APH)}{(21 - O_2\%\ in\ the\ gas\ leaving\ APH)} \times 100$$

The actual values may be compared against design and reasons for deviation if any, needs to be analysed.

11.8 Draft System

Thermal power plant has several fans such as Induced draft (ID) fans, Forced draft (FD) Fans, Primary air fans (PA fans). These fans contribute to significant auxiliary power consumption.

Performance evaluation of fans

Performance evaluation of FD, PA and ID fans can be assessed as per Chapter – 4.6 (Performance evaluation of fans and blowers).

The actual values should be compared with the design / Performance Guarantee (PG) test values and if any deviation is found the reasons for the same has to be investigated. Based on the actual operating parameters and diagnosis suitable energy saving measures can be implemented. The measures could include

- Replacement of fans
- Impeller replacement
- Variable speed drive application, etc

Study of air infiltration into the draft system

Air infiltration in the system has very adverse impact on the boiler loading, efficiency, power consumption of the ID fans, plant load factor etc.

Checks for air infiltration in to the system have to be performed periodically once in a month by monitoring oxygen content at the following sections:

- Before and after air preheater
- Before and after ESP
- Before and after ID fan

The difference in the oxygen gives the extent of air filtration into the system. Measurement of oxygen content across all the units in flue gas path, indicates the locations where infiltration is occurring.

Based on the oxygen content, coal flow and stoichiometric air requirement (in case where measurement of air flow across all units in the flue gas path is not possible) the flue gas quantity shall be estimated.

These gas quantities should be compared with the design/PG test or best-run values for that particular loading.

The induced draft effect is highest at ID fan inlet and the air in leakages can be pronounced in locations like ESP hoppers APH hoppers, duct work etc which are close to ID fans. Reduction in air infiltration will result in

- Reduced power consumption of ID fans
- Reduced boiler losses
- Improvement in boiler loading
- Increased unit load
- Increased capacity margins in ID fan

11.9 Water Pumping System

Water pumping is a vital energy consuming area in thermal power plant and the major pumps in thermal power plant are Boiler feed water pumps, Condensate extraction pumps, cooling tower pumps, raw water pumps, ash slurry pumps etc.

Boiler feed pump is the major consumer among all power consuming equipment in the power plant. BFP may constitute more than 20% of the total auxiliary power consumption.

For performance assessment of pumping system refer Chapter – 4.7 Energy performance assessments of water pumps

The actual values should be compared with the design / performance guarantee values and if any deviation is found the reasons for the same has to be investigated. Based on the actual operating parameters and diagnosis suitable energy saving measures can be implemented. The measures could include

- Replacement of pumps
- Impeller replacement/trimming
- Variable speed drive application
- Optimizing number of pumps in operation, etc

11.10 LP and HP Heaters

LP heaters and HP heaters constitute the regenerative portion of the Rankine cycle, wherein the temperature of the condensate from the hot well is boosted to, optimally higher values, upto the deaerator, in LP heaters, using steam extracted from the LP and IP turbines. From the deaerator onward, the fluid is referred to as feed water, which is pumped by the Boiler Feed Pumps (BFP's), through the HP heaters and then through the economizer into the boiler drum (Figure 11.7).

The performance of the feed water heaters can be analysed by assessing the following

Terminal Temperature Difference (TTD)

It is the difference between the saturation temperature at the operating pressure of the inlet steam to the heater and the temperature of the feed water leaving the heater (Figure 11.7).

Drain Cooler Approach (DCA)

It is the difference between the temperature of the drains (steam condensate) leaving the heater and temperature of the feed water entering the heater (Figure 11.7).

The lower the TTD and DCA, the more efficient the cycle. The more efficient cycle results in a lower heat rate and reduced fuel cost.

The illustrative profile of typical heat transfer zones in heaters is presented in Figure 11.7.

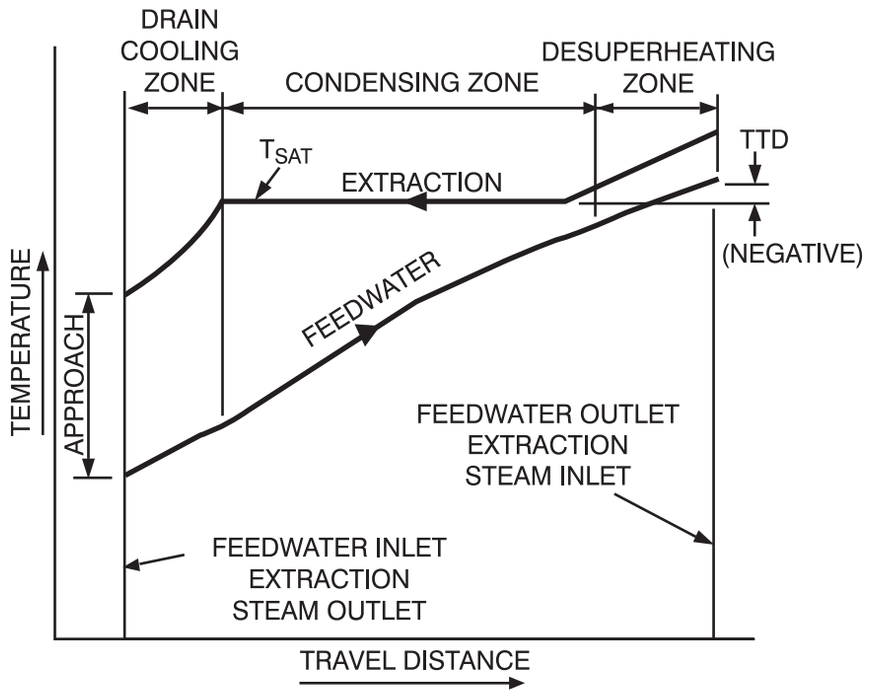


Figure: 11.7 Temperature profile for a feed water heater

In some of the cases, because of the de-superheating zone in the heater, the feed water temperature leaving the heater may be higher than the saturation temperature of the condensing zone. Therefore, the heater may have a negative TTD as shown in the figure. If the de-superheating zone of the heater is removed, the feed water outlet temperatures will be less than the saturation temperature, which results in a positive TTD. The practical lower TTD limit on a heater without a de-superheating zone is + 1.1 ^{o}C. The negative TTD limit for a heater with de-superheating zone depends on the amounts of superheat in the extraction steam entering the heater

Example 11.4. Calculation of performance parameters based on the data given in the following table.

Parameter Reference	UNIT	LP Heater
Extraction steam pressure	kg/cm^2 (a)	0.614
Extraction steam temperature	^{O}C	86.5
Shell steam pressure	kg/cm^2 (a)	0.488
Shell steam temperature	^{O}C	86.5
Saturated temperature of steam @ shell pressure	^{O}C	86.5
Inlet feedwater temperature	^{O}C	47
Outlet feedwater temperature	^{O}C	77.5
Drain outlet temperature	^{O}C	61.8

Terminal Temperature Difference (TTD)

TTD = Saturation temp. of extraction steam, ^{o}C – Feed water outlet temp., ^{o}C
TTD = 86.5 ^{o}C – 77.5 ^{o}C
TTD = 9 ^{o}C

Drain Cooler Approach (DCA)

DCA = Drain outlet temp., ^{o}C – Inlet feed water temp., ^{o}C
DCA = 61.8 ^{o}C – 47 ^{o}C
DCA = 14.8 ^{o}C

Increased values of TTD and DCA with respect to the design values indicate extent of drop in heat transfer mainly due to fouling.

11.11 Turbine

The steam turbines are split into three separate stages, High Pressure (HP), Intermediate Pressure (IP) and Low Pressure (LP) stage. After the steam has passed through the HP stage, it is returned to the boiler to be re-heated to its original temperature although the pressure remains greatly reduced. The reheated steam then passes through the IP stage and finally to the LP stage of the turbine.

ASME PTC-6S Steam Turbine performance test code is referred for assessing the performance of Steam Turbine.

The performance of the steam turbines can be analysed by assessing the following

- Turbine heat rate
- Turbine cycle efficiency (thermal efficiency)
- Stage (isentropic) turbine efficiency
- Specific steam consumption

Performance measurements

The temperature, pressure and flow measurements for the following are necessary

- Feed water at Inlet & Outlet of Heaters
- Main steam
- HP turbine extraction
- Hot reheat steam, Cold reheat Steam
- IP extraction
- IP Exhaust

In addition to the above, the generator output is to be noted

Turbine heat rate

Turbine heat rate is defined as the amount of heat input to the turbine in kcal for generating one unit of electricity.

$$\textbf{Turbine heat rate, kCal/kWh} = \frac{\mathbf{Q_1 \times (H_1 - h_2) + Q_2 \times (H_3 - H_2)}}{\textbf{Gross Generator Output}}$$

Q_1 - Main steam flow, kg/hr
H_1 - Main steam enthalpy, kcal/kg
h_2 - Feed water enthalpy, kcal/kg
Q_2 - Reheat steam flow, kg/hr
H_3 - Hot reheat enthalpy, kcal/kg
H_2 - Cold reheat enthalpy, kcal/kg

Unit heat rate

$$\textbf{Unit heat rate, kCal/kWh} = \frac{\textbf{Turbine heat rate, kCal/kWh}}{\textbf{Boiler efficiency}}$$

Example 11.5. Calculation of performance parameters based on the following data (refer figure 11.4)

Main steam pressure and temperature	: 150 kg/cm^2 (g), 535 ^{o}C
Main steam flow rate	: 627 TPH
Enthalpy of main steam	: 814.8 kcal/kg
Feed water temperature	: 244 ^{o}C
Feed water enthalpy	: 244 kcal/kg
Cold reheat steam pressure and temperature	: 36 kg/cm^2(g), 336 ^{o}C
Enthalpy of cold reheat steam	: 733.3 kcal/kg
Hot reheat steam pressure and temperature	: 35 kg/cm^2(g), 535 ^{o}C
Enthalpy of hot reheat steam	: 844.3 kcal/kg
Reheat steam flow	: 565 TPH
Generator output	: 210 MW
Boiler efficiency	: 87%

$$\text{Turbine heat rate, kCal/kWh} = \frac{627000 \times (814.8 - 244) + 565000 \times (844.3 - 733.3)}{210000}$$

$$\text{Turbine heat rate} = 2003 \text{ kCal/kWh}$$

$$\text{Unit heat rate} = \frac{2003}{0.87} = 2302 \text{ kCal/kWh}$$

Turbine cycle efficiency (thermal efficiency)

Turbine cycle efficiency is defined as the amount of electricity produced to the heat input to the turbine. It is reciprocal of heat rate in consistent units.

$$\textbf{Turbine cycle efficiency, \%} = \frac{\textbf{860}}{\textbf{Turbine heat rate}} \times \textbf{100}$$

$$\text{Turbine cycle efficiency, \%} = \frac{860}{2003} \times 100$$

$$\text{Turbine cycle efficiency} = 42.9\ \%$$

Turbine stage (isentropic) efficiency

The efficiency method given in this procedure is the enthalpy drop efficiency method. This method determines the ratio of actual enthalpy drop across turbine section to the isentropic enthalpy drop. This method provides a good measure for monitoring purposes.

Each section of the turbine must be considered as a separate turbine. Each section should be tested and the results are trended separately. While conducting the tests, it has to be ensured that, it is conducted over normal operating load range.

$$\textbf{Turbine stage (isentropic) Efficiency, \%} = \frac{\textbf{Actual enthalpy drop across the turbine}}{\textbf{Isentropic enthalpy drop across the turbine}} \times \textbf{100}$$

After evaluating the turbine heat rate and efficiency, the deviation from the design, if any should be assessed and the factors contributing to the deviations must be identified. The major factors to be looked into are:

- Main steam and reheat steam inlet parameters
- Turbine exhaust steam parameters
- Reheater and super heater spray
- Passing of high energy draining
- Loading on the turbine
- Boiler loading and boiler performance
- Operations and maintenance constraints
- Condenser performance and cooling water parameters
- Silica deposition and its impact on the turbine efficiency
- Inter stage sealing, balance drum and gland sealing
- Nozzle blocks
- Turbine blade erosion
- Functioning of the valves
- Operational status of HP heaters
- Performance of reheaters

Example 11.6.

A small scale biomass power plant is to be installed with the following parameters

1. Turbine inlet steam pressure is 66 kg/cm^2 (g) and temperature is 480 ^{o}C
2. Condensing pressure of the turbine is 0.18 kg/cm^2 (a)
3. Stage (isentropic) efficiency = 85%
4. Dryness factor of steam more than 88%
5. Boiler efficiency (thermal) = 71% on GCV of bagasse
6. GCV of fuel (bagasse) = 2276 kcal/kg
7. Maximum steam flow rate of Boiler = 80 TPH
8. Mechanical efficiency of turbine = 0.985
9. Efficiency of generator = 0.95
10. T-G gear efficiency = 0.98

Calculate the following

1. Steam generated by one tonne of bagasse
2. Quantity of bagasse required for 80 TPH of steam generation
3. Specific steam consumption of turbine
4. Power generated

Solution

Enthalpy of steam at 66 kg/cm^2 (g) and temperature 480 ^{o}C (H_1)

= 804.4 kcal/kg

Isentropic enthalpy of steam at 0.18 kg/cm^2 (a) ($H_{2\text{-}is}$)

= 528.8 kcal/kg

Actual enthalpy of steam at 0.18 kg/cm^2 (a) (H_2)

$= H_1 - \{\eta_{is} \text{ X } (H_1 - H_{is})\}$
$= 804.4 - \{0.85 \text{ X } (804.4 - 528.8)\}$
$= 570.1$ kcal/kg

Actual enthalpy drop
$= H_1 - H_2$
$= 804.4 - 570.1$
$= 234.3$ kcal/kg

Dryness fraction
$H_2 = h_w + 564.6 \text{ X}$
$570.1 = 57.38 + 564.6 \text{ X}$

X = 0.90, i.e. dryness fraction is within the limit

Steam generated by one tonne of bagasse

= 0.71 X 2276 / (804.4 – 57.38)

= 2.16 T/T

Quantity of bagasse required for 80 TPH

= 80/2.16

= 37.03 T

Specific steam consumption of turbine

$= 860/ \{(H_1 - H_2) X \eta_{mec} X \eta_{gen} X \eta_{gear} \}$

η_{mec} = 0.985

η_{gen} = 0.95

η_{gear} = 0.98

Specific steam consumption of turbine

= 860/ {(804.4 – 570.1) X 0.917)

= 4.0 kg/kWh

Power generated by 80 TPH of steam

= 80/4.0

= 20.0 MW

11.12 Condenser

Condenser is the most important equipment of power plant cycle. It plays a vital role in determining the heat rate of Turbine. It ensures maintaining of vacuum and hence the back pressure of the Turbine at which exhaust steam from turbine condenses. Lower the condenser back pressure lower is the condensing temperature of steam and hence increase in turbine cycle efficiency. The condensate formed is reused as feed water in Boiler through the feed water heaters and deaerator.

Due to the low condenser back pressure (high condenser vacuum), the exhaust steam from Turbine expands to a greater extent resulting in increased availability of heat energy for converting into mechanical work in Turbine and hence more Generator output. The Turbine output varies with condenser back pressure. As the condenser back pressure increases, the Turbine output decreases because each unit mass of steam does less work on the turbine.

The exhaust part of the steam turbines is designed with specific value of condenser back pressure. As the condenser back pressure increases above design value, the steam temperature in the condenser increases resulting in reduction of power output and increase in Turbine cycle heat rate and unit heat rates. Hence it is important to monitor the performance of the condenser and ensure its design back pressure.

Purpose of the Condenser Performance Test

Condenser is designed for certain cooling water inlet temperature, thermal load and condenser back pressure. The performance of the condenser is expected to deteriorate over a period of time due to bad chemistry of cooling water maintained in it resulting in scale formation and tube fouling affecting the heat transfer badly. Low cooling water flow through the condenser tubes raises the exhaust steam temperature and thus the condenser back pressure. Air ingress into the condenser through leaky valves, pipe fittings and instrument tapings and improper functioning of Steam Jet Air Ejector will also contribute to increased condenser back pressure. Hence performance assessment of the condenser is periodically required to determine:

i. Condenser effectiveness
ii. Condenser heat load
iii. The terminal temperature difference (TTD)

The values achieved during performance assessment are compared with the design values or performance guaranteed values or the values recorded during first time commissioning of the plant and the reasons for deviations are analysed and actions are taken to improve the performance of the condenser to its rated value and there by maintaining the efficiency of the Turbine cycle.

Performance Terms and Definitions

1. Condenser cooling water inlet temperature, ^{0}C =Tcwi
   The cooling water temperature shall be sufficiently lower than saturation temperature of the steam to result in a reasonable value of TTD.

2. Condenser terminal temperature difference (TTD),^{0}C = Tsat –Tcwo
   Where,
   Tsat = Saturation temperature of steam corresponding to condenser back pressure,0C
   Tcwo = Condenser cooling water outlet temperature,^{0}C

3. Cooling water temperature rise, ^{0}C = (Tcwo - Tcwi)

4. Maximum possible cooling water temperature rise, ^{0}C
   = (Saturation temperature of steam - Cooling water inlet temperature)
   = (Tsat – Tcwi)

5. Condenser effectiveness, %
   = Actual cooling water temperature rise/Maximum possible cooling water temperature rise
   = [(Tcwo -Tcwi) / (Tsat – Tcwi)] x 100

6. Condenser heat load, kcal/hr = Q x Cp x ΔT
   Where,
   Q = Condenser cooling water flow, kg/hr
   Cp = Specific heat of cooling water, kcal/kg 0C
   ΔT = Temperature rise of cooling water,^{0}C

7. Condenser vacuum, mmHg
   = (Atmospheric pressure, mmHg – Condenser back pressure, mmHg)

The above performance terms are explained in Figure 11.8. The schematic of condenser in power Plant cycle is shown in Figure 11.9. The effect of cooling water inlet temperature on condenser back pressure is shown in Figure 11.10. From the figure it is clear that as the cooling water inlet temperature is increasing the condenser back pressure is also increasing and at the same time the condenser terminal temperature difference (TTD) as indicated in Figure 11.11. The higher TTD indicates less heat transfer is taking place inside the condenser. Figure 11.12 shows the effect of condenser back pressure on power plant cycle efficiency. As the condenser back pressure is increasing the power plant cycle efficiency is decreasing which means increased heat rate of the cycle. The increased unit heat rate will lead to increase in fuel consumption cost and hence increased cost of power generation. The effect of back pressure on heat rate is considerable even for small variation of 0.01 kg/cm^2(a) as seen in the cases of 210 MW and 500 MW units from Table 11.7. Hence it is important to monitor the performance of the condenser and to operate the power plant at its rated heat rate value.

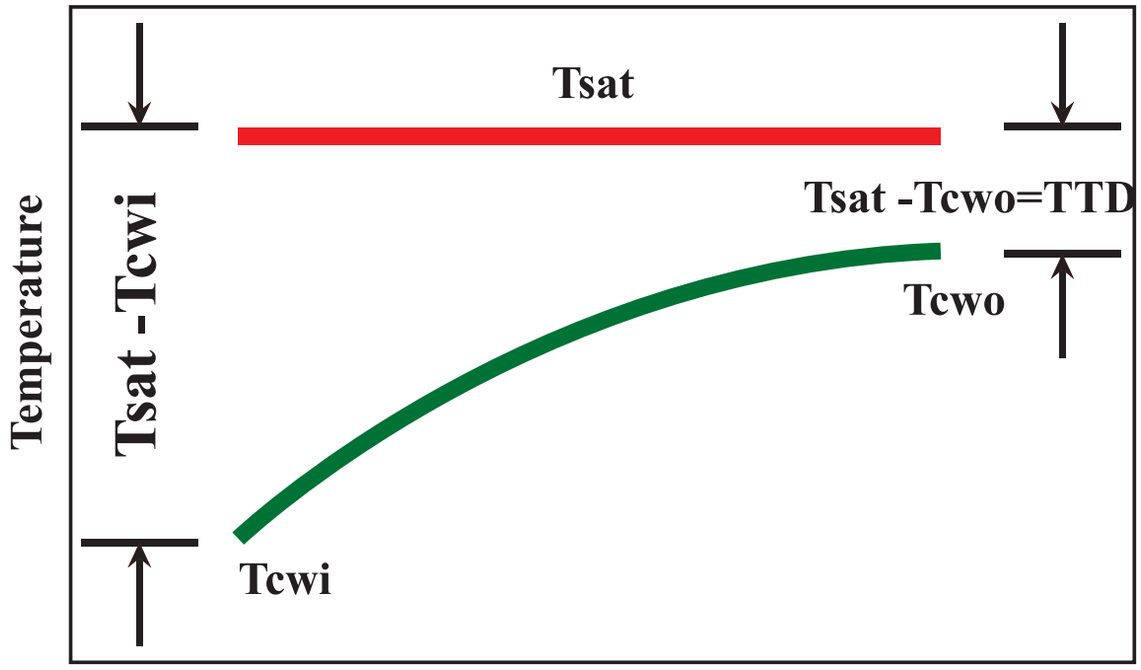


Figure 11.8 Temperature profile across condenser

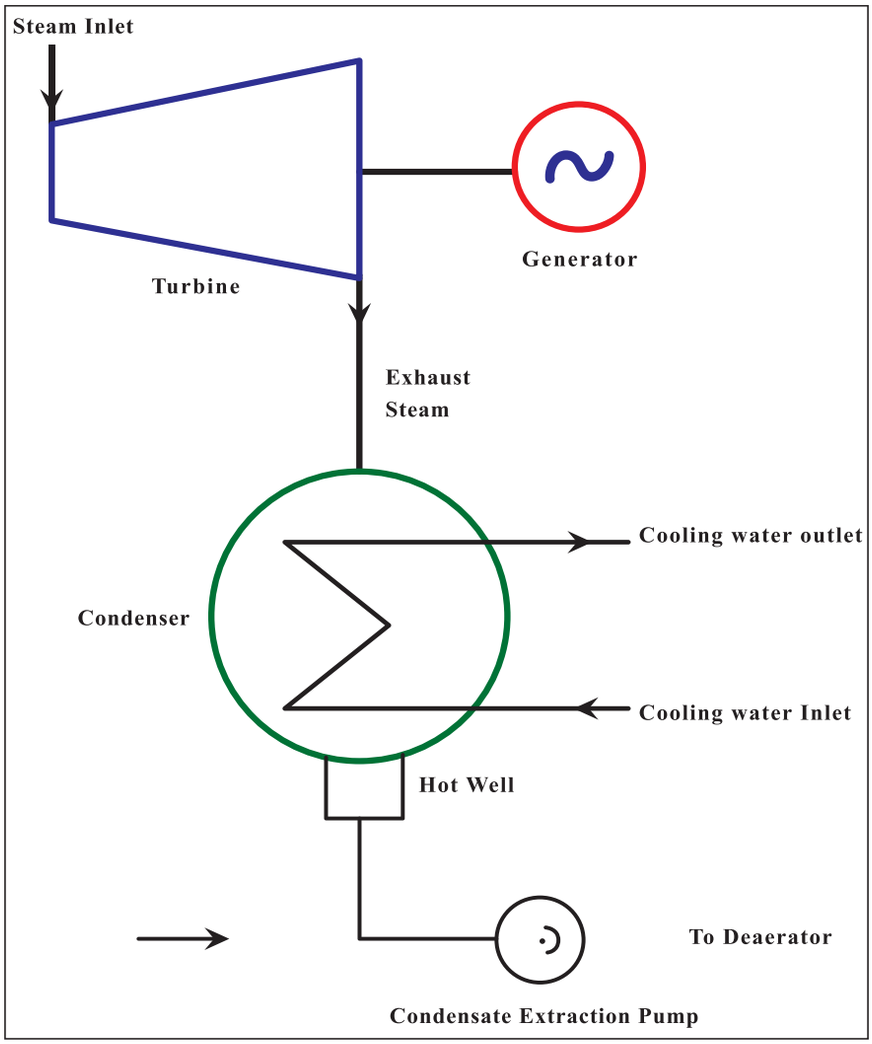


Figure: 11.9 Shell and Tube Surface Condenser

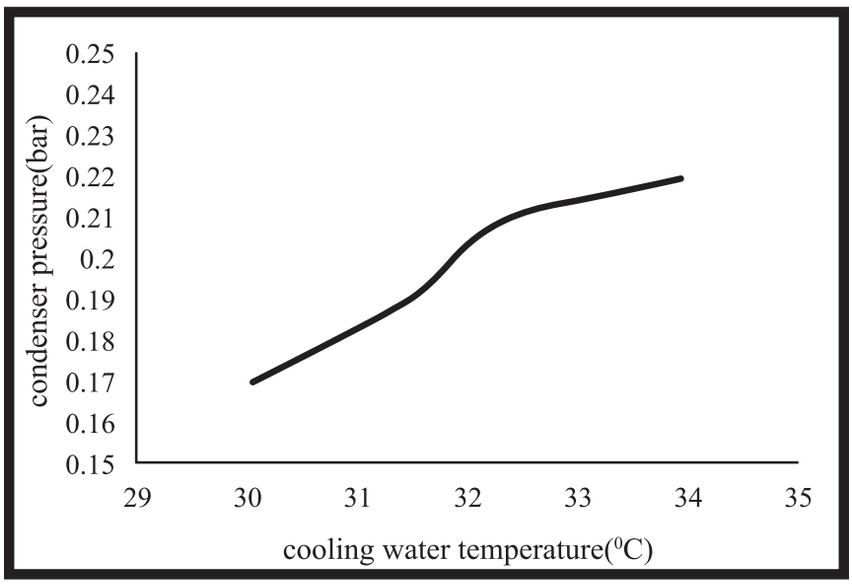


Figure: 11.10 Effect of cooling water inlet temperature on condenser back pressure

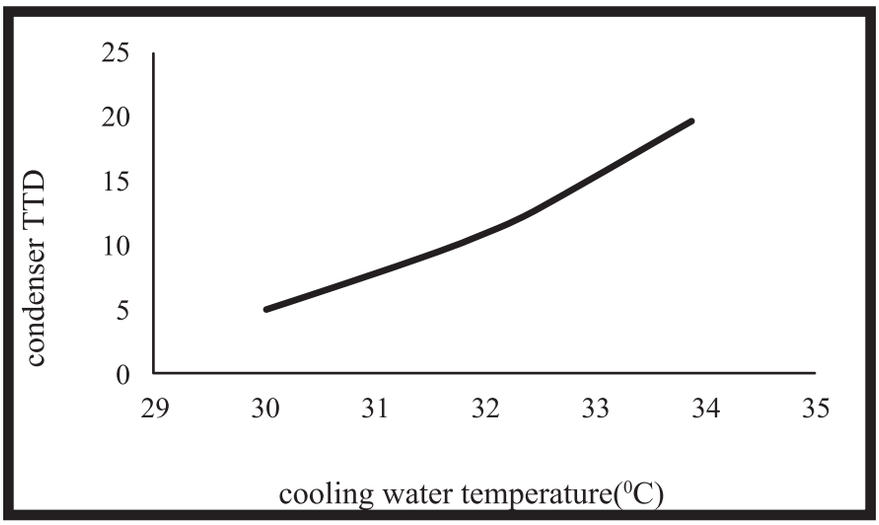


Figure: 11.11 Effect of cooling water inlet temperature on condenser TTD

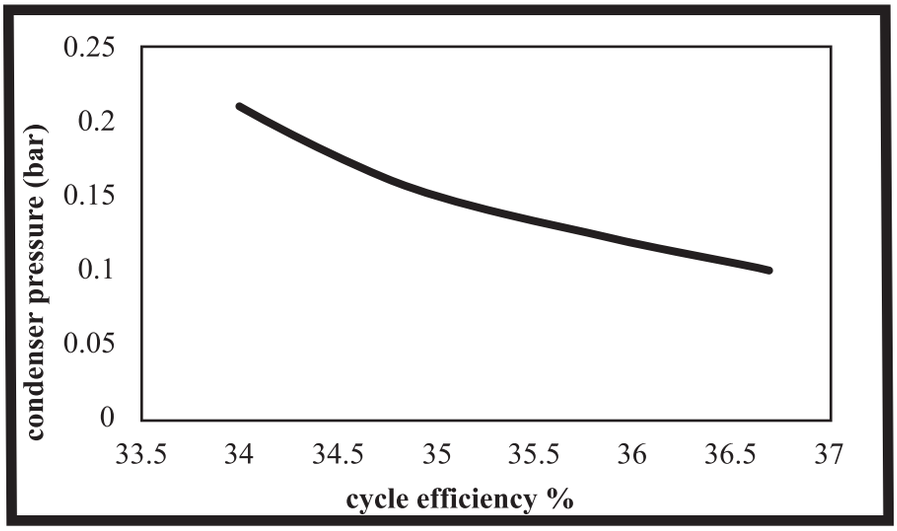


Figure: 11.12 Effect of condenser back pressure on power plant cycle efficiency

Table 11.7.Effect of condenser back pressure on heat rate of a typical turbine cycle

S. No.	Condenser Back Pressure	Heat Rate (kcal/kWh)	
	Ata	210 MW	500 MW
1.	0.07	1960	1975
2.	0.08	1968	1978
3.	0.09	1975	1980
4.	0.1	1990	1990
5.	0.11	2000	2000
6.	0.12	2012	2010
7.	0.13	2030	2020
8.	0.14	2050	2040

Performance Measurements

The following measurements are made at the condenser.

- Condenser cooling water inlet temperature
- Condenser cooling water outlet temperature
- Condenser back pressure/condenser vacuum
- Hot well condensate temperature
- Condenser cooling water flow

Factors Affecting Condenser Vacuum

The reasons for low condenser vacuum are:

- Reduced cooling water flow
- Increase in cooling water inlet temperature
- Fouling/scaling of condenser tubes
- Accumulation of gases
- Large thermal load
- Flooding of condenser tubes
- Improper functioning of Steam Jet Air Ejector

Condenser Vacuum Fault Diagnostics Chart

- High cooling water temperature rise (Tcwo – Tcwi)

  A rise in cooling water temperature is either due to low cooling water flow or large thermal load coming on the condenser. Cooling water flow being normal it represents high thermal load on the condenser.

- High terminal temperature difference (Tsat – Tcwo)

  An increase in terminal temperature difference indicates interference with heat transfer due to scaling/fouling of condenser tubes.

- High temperature difference between Tsat – Tc

  A rise in temperature difference between Tsat and condensate temperature Tc; and increase in dissolved oxygen concentration of condensate indicates air ingress into the condenser.

Example 11.7.

If the condenser back pressure is 0.1 ata, calculate the condenser vacuum. Given that the atmospheric pressure is 752 mmHg.

Solution:

Ata is the abbreviation for atmospheric pressure absolute and is defined as the standard atmospheric pressure at sea level.

1 ata = 1atm = 760mmHg = 1.0332kg/cm^2 = 1.01325bar = 101.3kPa = 10.33mH_2O = 29.92 inches of Hg = 14.7psia (Book-1, Chapter-3)

Given data: 0.1 ata = 760 x 0.1 = 76 mmHg

Atmospheric pressure = 752mmHg

Condenser vacuum, mmHg = (Atmospheric pressure, mmHg - Condenser back pressure, mmHg)

= (752 - 76) = 676 mmHg.

Example 11.8.

The exhaust steam pressure of a turbine is 0.051 kg/cm^2(a). Calculate the condenser vacuum in millibar. Assume atmospheric pressure as 760 mmHg.

Solution:

1.0332 kg/cm^2 (a)	=	760 mmHg
0.051 kg/cm^2 (a)	=	760 x 0.051/1.0332 = 37.51 mmHg
Condenser vacuum	=	760 - 37.51 = 722.49 mmHg
722.49 mmHg	=	1.01325 x 722.49/760 = 963.24 millibar
Condenser vacuum	=	963.24 millibar

Example 11.9.

If the condenser back pressure of a turbine is 0.11 bar(a), what is its condenser vacuum in millibar ? Atmospheric pressure is 755 mmHg.

Solution:

1.01325 bar (a)	=	760 mmHg
0.11 bar (a)	=	760 x 0.11/1.01325 = 82.5 mmHg
Condenser vacuum	=	755 - 82.5 = 672.5 mmHg
Condenser vacuum	=	1.01325 x 672.5/760 = 896.59 millibar

Example 11.10.

The vacuum recorded in a steam power plant is 700 mmHg. Find out the absolute pressure inside the condenser in kg/cm^2 if the atmospheric pressure is 740 mmHg.

Solution:

Absolute pressure in condenser = (Atmospheric pressure - Condenser vacuum)

Absolute pressure = 740 – 700 = 40 mmHg
760 mmHg = 1.0332 kg/cm^2 (a)
40 mmHg = 1.0332 x 40/760 = 0.0544 kg/cm^2 (a)
Condenser pressure = 0.0544 kg/cm^2 (a)

Example 11.11.

The cooling water inlet temperature and outlet temperature of a condenser is 30^{0}C and 39^{0}C respectively. If the condenser vacuum is 684 mmHg and the atmospheric pressure is 750 mmHg. Calculate the condenser effectiveness and TTD.

Solution:

(i) Calculation of condenser effectiveness

Condenser vacuum = 684 mmHg
Atmospheric pressure = 750 mmHg
Condenser back pressure = (750 - 684) = 66 mmHg
We know, 760 mmHg = 1.0332 kg/cm^2
66 mmHg=(1.0332 x 66)/760 = 0.089725 kg/cm^2

From steam tables, at 0.089725 kg/cm^2, Saturation temperature of steam = 43.3^{0}C
Condenser effectiveness = (Tcwo - Tcwi) / (Ts - Tcwi) x 100
= (39 -30)/ (43.3—30) = 67.7%

(ii) Calculation of Condenser TTD
TTD = (Ts - Tcwo) = 43.3 – 39 = 4.3^{0}C

Example 11.12.

The condenser of a 210 MW turbine is designed for the following conditions.

Turbine exhaust steam flow to condenser = 560 T/hr
Condenser back pressure = 0.1042 bar(a)
Dryness fraction of steam = 88%
Cooling water inlet temperature = 30^{0}C
Cooling water outlet temperature = 40^{0}C

Calculate:
(i) Heat load on condenser
(ii) Cooling water flow required to condense the steam

Solution:

(i) Heat load calculation:

Heat load = Steam inlet flow x L.H. of vaporisation of steam

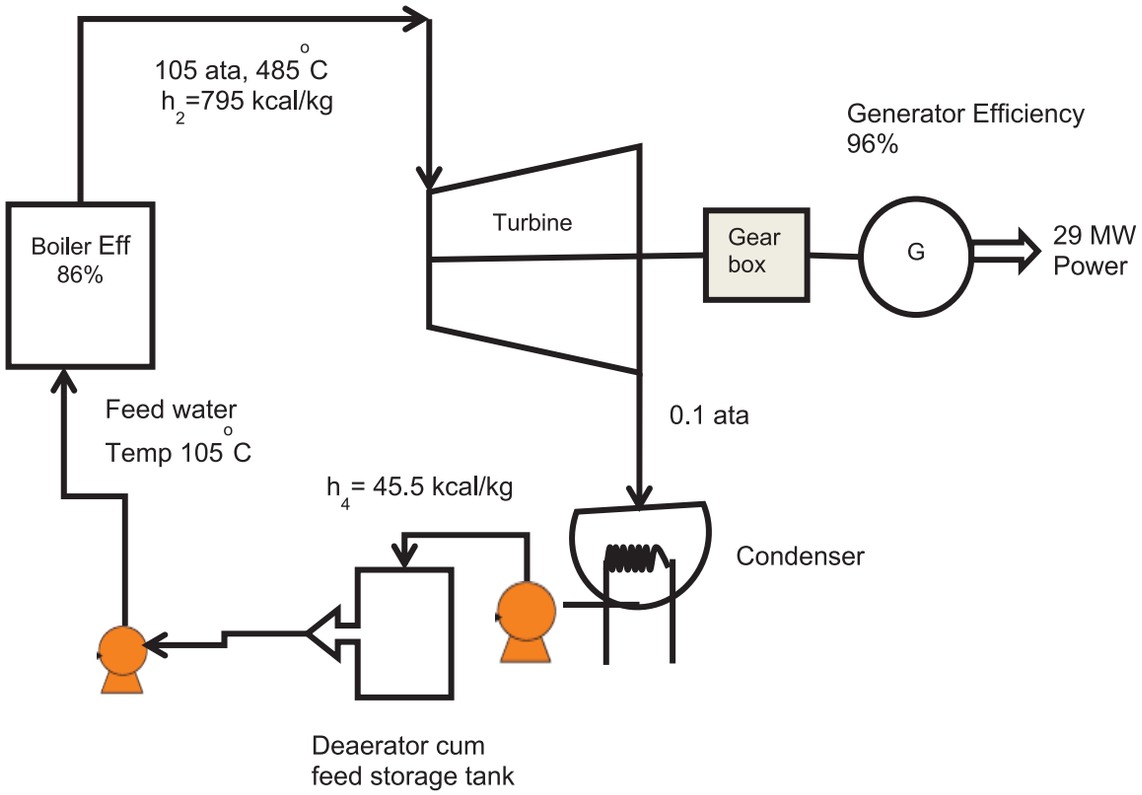
From steam tables at 0.1042 bar (a), L.H. of vaporisation of steam is 2391 kJ/kg
L.H. of 88% dry steam = 2391 x 0.88=2104 kJ/kg
Heat load = 560 x 1000 x 2104 = 1178240000 kJ/hr

(ii) Calculation of cooling water flow Q:

Heat removed by cooling water = Q x Cp x (Tcwo - Tcwi)
1178240000 / 4.18 = Q x 1 x (40 - 30)
Q= 28187.6 T/hr

Solved Example:

A captive thermal plant is delivering an output of 29 MW at the generator terminal. The generator efficiency is 96%. The steam generated in a utility boiler with an efficiency of 86% at 105 ata and 485°C is fed to the turbine. The turbine exhausts steam to condenser maintained at 0.1 ata and 45.5°C. The feed water temperature at inlet to the boiler is 105°C.



The other data pertaining to captive power plant are,

Enthalpy of steam at 105 ata 485°C	= 795 kcal/kg.
Dryness fraction of steam at inlet to condenser	= 0.9
Enthalpy of dry saturated steam at 0.1ata	= 618 kcal/Kg.
Enthalpy of water at 0.1 ata & at 45.5°C	= 45.5 kcal/Kg.
Loss in the gear box connecting turbine and generator	= 1100 kW
Enthalpy of feed water at inlet to the boiler ,	= 105 kcal/Kg.

Based on the above data determine:

i. Output of the steam turbine in kW
ii. Steam flow through the turbine
iii. Turbine heat rate
iv. Unit heat rate

Solution:

Enthalpy of steam at turbine exhaust = 45.5 +0.9 (618 – 45.5)

i.e. h_3 = 560.75 Kcal/Kg.

Generator electric output = 29000 KW
Generator input = 29000 / 0.96 = 30208.33 KW

Loss in gear box = 1100 KW
Output of steam turbine = Generator input + Gear box loss
= 30208.33 + 1100
= 31308.33 kW

i) Output of the steam turbine **Say = 31308 kW**

ms = Steam flow through turbine

$$= \frac{\text{(Turbine output x 860)}}{(h_2 - h_3)\ \text{Turbine enthalpy drop}}$$

h_2 = Enthalpy at turbine inlet = 795 kcal/kg
h_3 = Enthalpy at turbine exhaust = 560.75 kcal/kg.
m_s = 31308 x 860 / 795 – 560.75) = 114940.78 kg/hr
= 114.94 TPH

ii) Steam flow through the turbine **Say = 115 TPH**

iii) Turbine heat rate = Heat input to turbine / Generator output
= $[m_s (h_2 - h_1)] / 29000$
= 115000 (795 – 105) / 29000
= 2736.2 kcal/ kWh

iv) Unit heat rate = Turbine heat rate / Efficiency of boiler
Unit heat rate = 2736.2 / 0.86 = **3181.63 kcal/ kWh**

QUESTIONS	
S-1	Explain the difference between gross heat rate and net heat rate.
S-2	What is meant by reheat? How does it improve the efficiency of the thermal power plant?
S-3	What do you understand by regeneration?
S-4	What is the purpose of the condensate extraction pump?
S-5	What is meant by air to coal ratio?
S-6	What should be the ideal coal fineness for a power plant boiler?
S-7	Define terminal temperature difference
S-8	How are heat rate and efficiency of the power plant related?
S-9	What is the purpose of primary air in the power plant?
S-10	If the heat of a power plant is 2900 kcal/kWh. What is its efficiency?
L-1	Explain how TTD can also be negative
L-2	Explain how air infiltration can be estimated in the flue gas path.
N-1	A performance evaluation of a PA fan resulted in the following data. Pitot tube measurement average Velocity pressure : 75 mm water column Suction pressure : - 20 mm water column Outlet pressure : 480 mm water column Area of duct : 8 square meter Pitot tube constant : 0.85 Corrected gas density : 1.15 kg/ m^3 (i) Calculate flow in m^3/sec. (ii) Calculate the static fan efficiency based on the following 3 phase motor data. Line current : 100 Amps Line voltage : 11,000 volts Power factor of electric motor : 0.9 Efficiency of motor at the operating load is 95%.

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Chapter 12

Performance Assessment of Steel Sector

Learning Objectives

In this chapter you will learn about

- Iron and steel making routes
- Primary and secondary steel making process
- Energy interaction in steel plants
- Case example – Heat balance of rotary hearth furnace in a secondary steel plant

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Understand the process of steel making in primary and secondary steel sectors**
- ✓ **Perform heat balance calculation of a rotary hearth furnace**

12. ENERGY PERFORMANCE ASSESSMENT OF STEEL SECTOR

12.1 Introduction

Iron and steel is the largest consumer of energy among all industrial sectors. The sector currently consumes about 10% of total electricity and 27% of coal used by Indian industry. The energy cost contributes to nearly 30 – 35% of this sector's production cost. In a typical integrated steel plant the coking coal is the predominant source of primary energy. It has long been an established fuel economy policy for integrated steel mills to try their best in utilizing by-product gases derived from iron- and steelmaking, while limiting purchase of fuels and electric power from outside. Some 40% is accounted for by the byproduct gases.

12.2 Iron and Steel Making Routes

Currently there are two main routes for the production of steel: Production of primary steel using iron ores and scraps and production of secondary steel using sponge iron and scraps (Figure 12.1). A wide variety of steel products are produced by the industry, ranging from slabs and ingots to thin sheets, which are used in turn by a large number of other manufacturing industries.

Primary steel: Pig iron, the input for primary steel production is produced in a blast furnace, using coke in combination with injected coal, to reduce sintered or pelletized iron ore to pig iron. Limestone is added as a fluxing agent. Coke is produced in coke ovens. Reduction of the iron ore is the largest energy-consuming process in the production of primary steel. Besides iron, the blast furnace also produces blast furnace gas (used for heating purposes), electricity (if top gas pressure recovery turbines are installed) and slags (used as building materials). Further steel is produced by two processes: open hearth furnace (OHF) and basic oxygen furnace (BOF).

While OHF uses more energy, this process can also use more scrap than the BOF process. However, BOF process is rapidly replacing OHF worldwide, because of its greater productivity and lower capital costs. In addition, this process needs no net input of energy and can even be a net energy exporter in the form of BOF-gas and steam. The process operates through the injection of oxygen, oxidizing the carbon in the hot metal. Several configurations exist depending on the way the oxygen is injected. The steel quality can be improved further by ladle refining processes used in the steel mill.

Secondary steel: Sponge iron and Scrap are melted and refined, using a strong electric current, AC or DC, in an electric arc furnace (EAF) to produce steel. DRI is produced by reduction of the ores below the melting point and has different properties than pig iron.

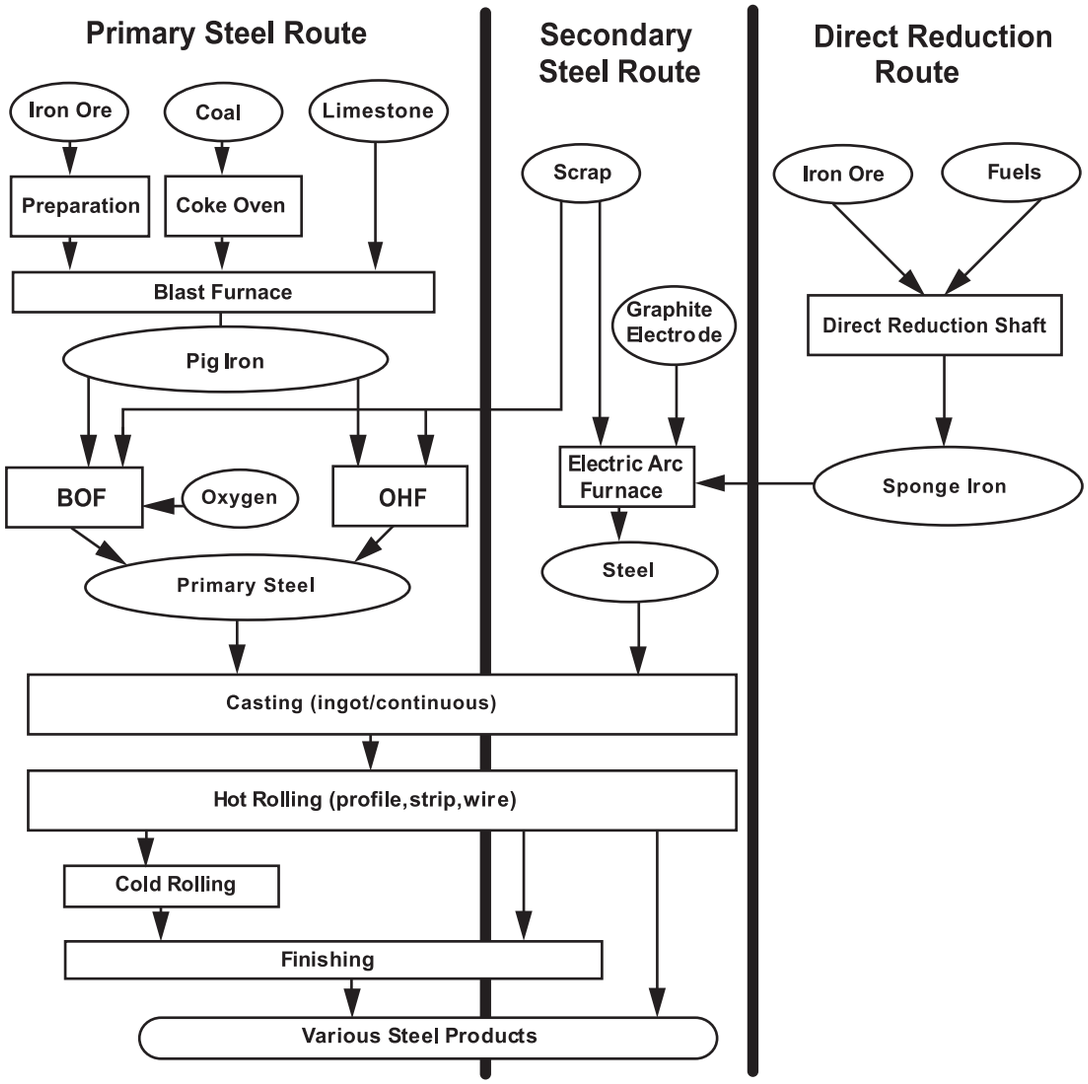

Figure 12.1 Different process routes to steel making

12.3 Primary Steel Process

Steel production at an integrated steel plant using the primary process involves the following four basic steps i.e,

- Production of coke and sinter / pallets from iron fines - Material preparation
- Reduction of iron ore in blast furnace-Iron making Figure 12.2 represents the process flow chart of an integrated steel plant.
- Processing of molten iron to produce steel (Steel making)
- Steel forming and finishing.

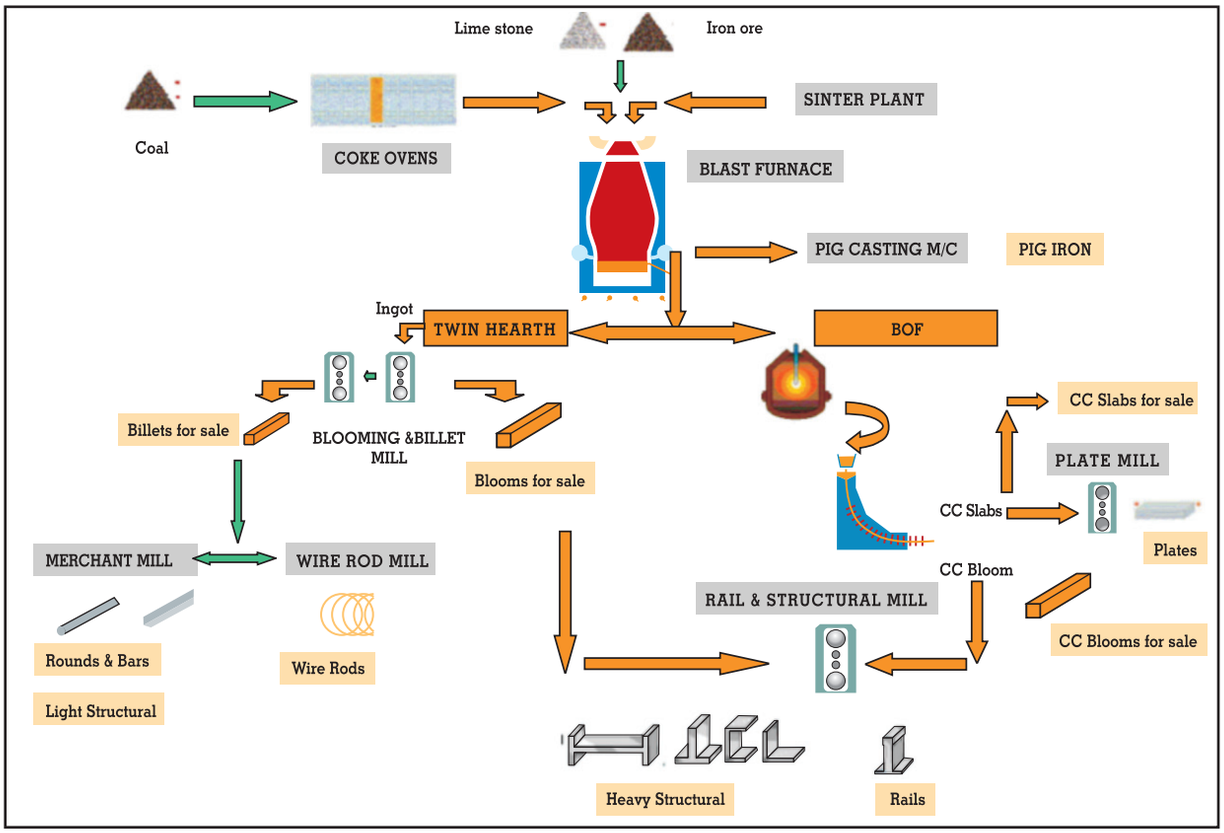

Figure 12.2 Process Flow Chart of an Integrated Steel Plant

Coke ovens-sinter-BF-BOF route

The most common steel making technology is the BF-BOF route. Coke is used in Blast Furnace (BF) both as a reductant and as a source of thermal energy. It involves reduction of ore to liquid metal in the blast furnace and refining in convertor to form steel. The various stages of the steel plant is described below.

Coke making - coal carbonisation: In the coke-making process, bituminous coal is fed (usually after processing operations to control the size and quality of the feed) into a series of ovens, which are sealed and heated at high temperatures in the absence of oxygen. Volatile compounds that are driven off the coal are collected and processed to recover combustible gases and other by-products. The solid carbon remaining in the oven is coke. It is taken to the quench tower, where it is cooled with a water spray or by circulating an inert gas, a process known as dry quenching. The coke is screened and sent to a blast furnace or to storage. A schematic diagram of Coke Oven Battery is given in Figure 12.3.

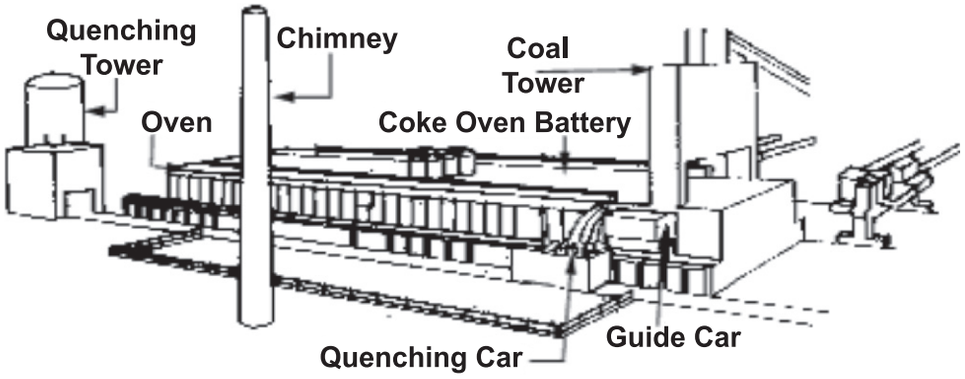

Figure 12.3: Coke Oven Battery

The coal-to-coke transformation takes place as follows: The heat is transferred from the heated brick walls into the coal charge. From about 375 °C to 475 °C, the coal decomposes to form plastic layers near each wall. At about 475 °C to 600 °C, there is a marked evolution of tar, and aromatic hydrocarbon compounds, followed by resolidification of the plastic mass into semi-coke. At 600 °C to 1100 °C, the coke stabilization phase begins. This is characterized by contraction of coke mass, structural development of coke and final hydrogen evolution. During the plastic stage, the plastic layers move from each wall towards the center of the oven trapping the liberated gas and creating in gas pressure build up which is transferred to the heating wall. Once, the plastic layers have met at the center of the oven, the entire mass has been carbonized. The incandescent coke mass is pushed from the oven and is wet or dry quenched prior to its shipment to the blast furnace.

Sintering

Sintering is a technology for agglomeration of iron ore fines into useful Blast Furnace burden material. This technology was developed for the treatment of the waste fines in the early 20th century. Since then sinter has become the widely accepted and preferred Blast Furnace burden material.

The major advantages of using sinter in Blast Furnaces are :

- Use of iron ore fines, coke breeze, metallurgical wastes, lime, dolomite for hot metal production
- Better reducibility and other high temperature properties
- Increased BF productivity
- Improved quality of hot metal
- Reduction in coke rate in blast furnaces

A Sinter plant typically comprise the following sub-units as shown in Figure 12.4.

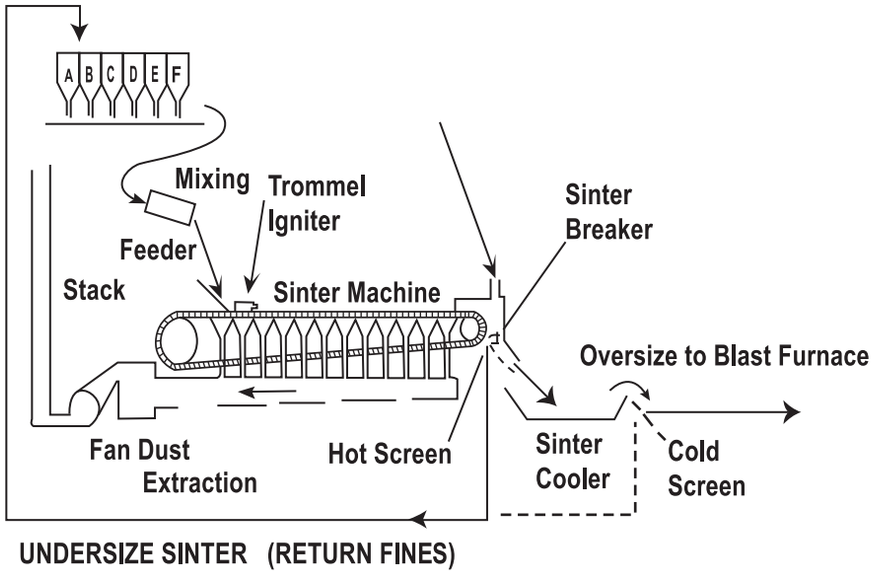

Figure 12.4 Sintering Process

The raw materials used are as follows - Iron ore fines (-10 mm), coke breeze (-3 mm), Lime stone & dolomite fines (-3mm) and other metallurgical wastes. The proportioned raw materials are mixed and moistened in a mixing drum. The mix is loaded on sinter machine through a feeder onto a moving grate (pallet) and then the mix is rolled through segregation plate so that the coarse materials settle at the bottom and fines onto the top.

The top surface of the mix is ignited through stationary burners at 1200 ^{o}C. As the pallet moves forward, the air is sucked through wind box situated under the grate. A high temperature combustion zone is created in the charge -bed due to combustion of solid fuel of the mix and regeneration of heat of incandescent sinter and outgoing gases. Due to forward movement of pallet, the sintering process travels vertically down.

Sinter is produced as a combined result of locally limited melting, grain boundary diffusion and recrystallisation of iron oxides.

On the completion of sintering process, finished sinter cake is crushed and cooled. The cooled sinter is screened and + 6 mm fraction is despatched to blast furnace and -6 mm is recirculated as return sinter.

Blast Furnace

The Blast furnace iron making process (Figure 12.5) basically consists of the conversion of iron oxide to iron in liquid form. This requires reductant for reduction of iron oxide and heat for the reduction reaction to take place and melting the products of smelting. The primary source to fulfill both these requirements is carbon (in the form of coke), which shares major portion of cost of hot metal production.

The raw materials passes through numerous chemical and physical reactions while descending to the bottom of the furnace. The iron ore, pellets and sinter are reduced which simply means the oxygen in the iron oxides is removed by a series of chemical reactions. These reactions occur as follows:

1) $3\ Fe_2O_3 + CO = CO_2 + 2\ Fe_3O_4$	**Begins at 450 °C**
2) $Fe_3O_4 + CO = CO_2 + 3\ FeO$	**Begins at 600 °C**
3) $FeO + CO = CO_2 + Fe$ or $FeO + C = CO + Fe$	**Begins at 700 °C**

At the same time the iron oxides undergo these purifying reactions, they also begin to soften, melt and finally trickle as liquid iron through the coke to the bottom of the furnace.

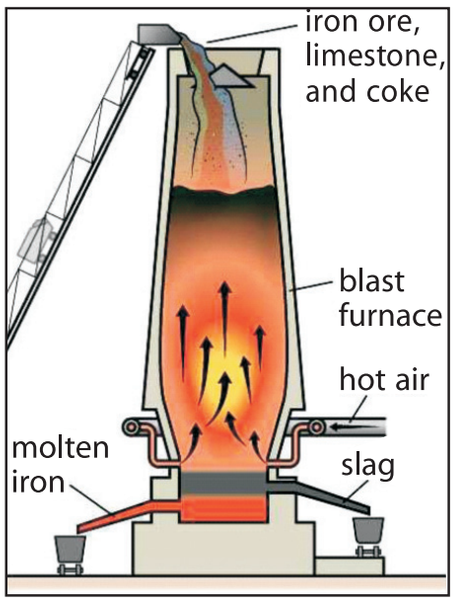

Figure 12.5 Blast Furnace

The coke descends to the bottom of the furnace to the level where the preheated air or hot blast enters the blast furnace. The coke is ignited by this hot blast and immediately reacts to generate heat as follows:

$C + O_2 = CO_2 + Heat$

Since the reaction takes place in the presence of excess carbon at a high temperature the carbon dioxide is reduced to carbon monoxide as follows:

$CO_2 + C = 2CO$

The product of this reaction, carbon monoxide, is necessary to reduce the iron ore as seen in the previous iron oxide reactions.

The limestone descends in the blast furnace and remains a solid while going through its first reaction as follows:

$CaCO_3 = CaO + CO_2$

This reaction requires energy and starts at about 875°C. The CaO formed from this reaction is used to remove sulphur from the iron which is necessary before the hot metal becomes steel. This sulphur removing reaction is:

$FeS + CaO + C = CaS + FeO + C$

The CaS becomes part of the slag. The slag is also formed from any remaining Silica (SiO_2), Alumina (Al_2O_3), Magnesia (MgO) or Calcia (CaO) that entered with the iron ore, pellets, sinter or coke. The liquid slag then trickles through the coke bed to the bottom of the furnace where it floats on top of the liquid iron since it is less dense.

Another product of the iron making process, in addition to molten iron and slag, is hot dirty gases. These gases exit the top of the blast furnace and proceed through gas cleaning equipment where particulate matter is removed from the gas and the gas is cooled. This gas has a considerable energy value so it is burned as a fuel in the "hot blast stoves" which are used to preheat the air entering the blast furnace to become "hot blast". Any of the gas not burned in the stoves is sent to the boiler house and is used to generate steam which turns a turbo blower that generates the compressed air known as "cold blast" that comes to the stoves.

In summary, the blast furnace is a counter-current reaction where solids descend and gases ascend. In this reaction there are numerous chemical and physical reactions that produce the desired final product which is hot metal.

Hot metal produced in the blast furnace is sent to Basic Oxygen Furnace (BOF) for steel making or to Pig casting machines for pig iron casting in ladles.

Basic Oxygen Furnace

The basic oxygen process is the most common process for producing steel. The basic oxygen furnace is a pear shaped vessel lined inside with refractory bricks. The vessel lining consists of tar bonded dolomite /magnesia carbon bricks or other refractories. The vessel can be rotated 360^{o} on its axis.

The 'heat' begins with the addition of scrap into the slightly tilted convertor, hot metal is then added after straightening the convertor. Oxygen is blown into the bath through the water cooled lance. The necessary fluxes are added during blowing. Flux addition is done automatically and precisely through bunkers situated above the convertors. A sample is taken after blowing for 16-18 minutes and temperature is measured using a thermocouple. The steel is tapped by tilting the convertor to the tapping side and alloying elements are added via chutes while metal is being tapped. The convertor is tilted to the charging side in order to remove the floating slag.

During blowing operation, oxygen oxidises iron into iron oxide and carbon into carbon monoxide. The iron oxide immediately transfers the oxygen to the tramp elements. The center of the reaction has temperatures of around 2000-2500 ^{o}C. The development of carbon monoxide during refining process promotes agitation within the molten bath. The reaction of the tramp elements with the oxygen and the iron oxide developed in the center of reaction leads to formation of reactive slag. As blowing continues, there is a continuous decrease of carbon, phosphorous, manganese and silicon within the melt. Phosphorous is removed by inducing early slag formation by adding powder lime with oxygen. The refining process is completed when the desired carbon content is attained.

Continuous Casting (CC)

Continuous casting (Figure 12.6) technique accounts for more than 60% of total liquid steel in the world. The main advantages of steel processing through CC route are higher yield, lower energy consumption, elimination of primary mills.

During continuous casting, the liquid steel passes from the pouring ladle, with the exclusion of air, via a tundish with an adjustable discharge device into the short, water-cooled copper mould. The shape of the mould defines the shape of the steel. Before casting, the bottom of the mould is sealed with a so-called dummy bar. As soon as the bath reaches its intended steel level, the mould starts to oscillate vertically in order to prevent the strand adhering to its walls. The red-hot strand, solidified at the surface zones, is drawn from the mould, first with the aid of a dummy bar, and later by driving rolls. Because of its liquid core, the strand must be carefully sprayed and cooled down with water. Rolls on all sides must also support it until it has completely solidified. This prevents the still thin rim zone from disintegrating.

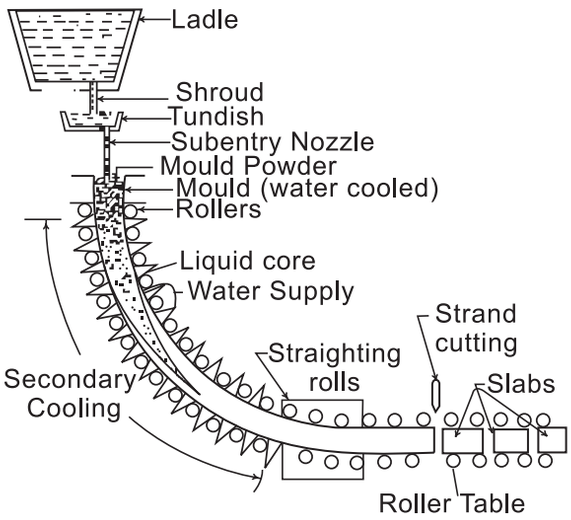

Figure 12.6 Continuous Casting

Once it has completely solidified, the strand can be divided by mobile cutting torches or shears. Intensive cooling leads to a homogeneous solidification microstructure with favourable technological properties. The cast material is sold as ingots or slabs to steel manufacturing industries. However, most of the steel is rolled by the steel industry to sheets, plates, tubes, profiles or wire.

Rolling and Finishing

Generally the steel is first treated in a hot rolling mill. The steel is heated and passed through heavy roller sections for reducing the thickness of the steel. Hot rolling produces profiles, sheets, or wire. After hot rolling, the sheets may be reduced in thickness by cold rolling. Finishing is the final production step, and may include different processes such as annealing, pickling, and surface treatment. A more

advanced technology, near-net-shape casting, reduces the need for hot rolling because products are cast closer to their final shape. (Figure 12.7).

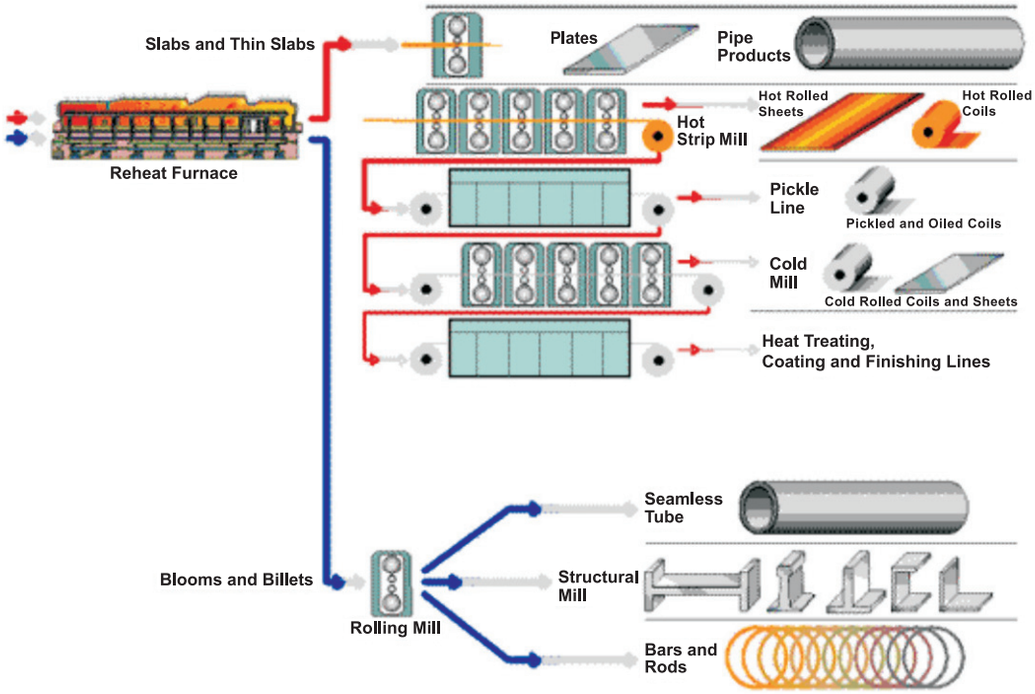

Figure 12.7 Rolling and Finishing

Energy Consumption

Iron & Steel industry in India is highly energy intensive. Major energy inputs in the sector are coking coal, non-coking coal, coke & electricity. The specific energy consumption in Indian Steel plants is quite high. It ranges between 6.17 Gcal/ tcs to 8 Gcal/ tcs (tonne of crude steel). There is variation of specific energy consumption in different steel plants. This is mainly because of different processes, quality of coal, types of products produced & energy efficiency measures adopted by the plants. The details of specific energy consumption by the Indian steel plants (Gcal/ tcs) are given in Table 12.1. below:-The specific energy consumption for Indian plants are much above the World level of 4.5 Gcal/ tcs.

Table 12.1 Specific energy consumption of Indian steel plants (Gcal/tcs)

Plant	2008-09
Bhilai Steel Plant (BSP)	6.50
Durgapur Steel Plant (DSP)	6.50
Rourkela Steel Plant	7.09
Bokaro Steel Plant (BSL)	6.73
IISCO Steel Plant (ISP)	8.27
Rashtriya Ispat Nigam Ltd. (RINL)	6.77
Tata Steel Ltd.	6.59

Source: Joint plant committee, Government of India

The major energy consuming process in iron making are coking, sinter making & blast furnace. They consume about 61.3% of the total energy. The slabbing mill, hot strip mill and cold rolling mill together with others account for 36.5% energy consumption. The table 12.2 gives the major portion of energy consumption in iron making.

Table 12.2 Energy consumption of different processes in a typical steel plant

Process	Energy consumed 10^6 kCal/tonne CS	% of total energy	
Coking	1.033	11.5	61.3% in iron making
Sinter making	0.967	10.7	
Blast furnace	3.519	39.1	
BOF (LD)	0.202	2.2	2.2% in steel making
Slabbing mill	0.483	5.4	36.5% in rolling and others
Hot strip mill	1.080	12.0	
Cold rolling mill	1.025	11.4	
Other (including losses)	0.91	7.7	
Total	**9.000**	**100.0**	

12.4 Secondary Steel Process

In the Electric Arc Furnace (EAF) (Figure – 12. 8) steelmaking process, the coke production, pig iron production, and steel production steps are omitted, resulting in much lower energy consumption. To produce EAF steel, scrap is melted and refined, using a strong electric current. Several process variations exist, using either AC or DC currents and fuels can be injected to reduce electricity use. The EAF is equipped with eccentric bottom tapping, ultra high power transformers, oxygen blowing, full foamy slag operation, oxy-fuel burners, and carbon injection.

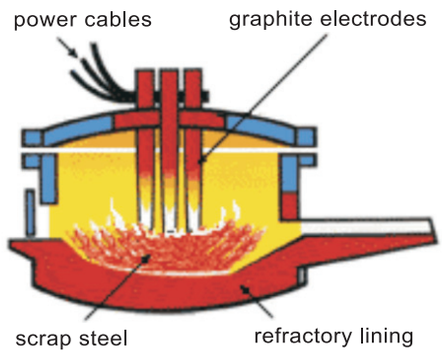

Figure 12.8 Electric Arc Furnace

Process

In a typical heat cycle, commonly referred to as the "tap-to-tap cycle", the cycle starts with the charging of the furnace with steel scrap. After the furnace is charged and the roof is in place, the operator lowers the electrodes, each of which has its own regulator and mechanical drive. Current is initiated and the electrodes bore through the scrap to form a pool of liquid metal. The scrap helps to protect the furnace lining from the high intensity arc during meltdown. Subsequently, the arc is lengthened by increasing the voltage to maximum power. Most modern furnaces are equipped with water-cooled panels in the upper half of the sidewall, rather than refractories, which allow for longer arcs and higher energy input

to the furnace. In the final stage, when there is a nearly complete metal pool, the arc is shortened to reduce radiation heat losses and to avoid refractory damage and hot spots. After meltdown, oxygen is injected to oxidize the carbon in the steel or the charged carbon. The decarburization process is an important source of energy. In addition, the carbon monoxide that evolves helps to flush nitrogen and hydrogen out of the metal. It also foams the slag, which helps to minimize heat loss and shields the arc-thereby reducing damage to refractories.

Energy balance

A typical energy balance (Sankey diagram) for a modern EAF is shown in Figure 12.9 & 12.10. Depending upon the melt shop operation, about 60 to 65% of the total energy is electric, the remainder being chemical energy arising from the oxidation of elements such as carbon, iron, and silicon and the burning of natural gas with oxy-fuel burners. About 53% of the total energy leaves the furnace in the liquid steel, while the remainder is lost to the slag, waste gas, and cooling.

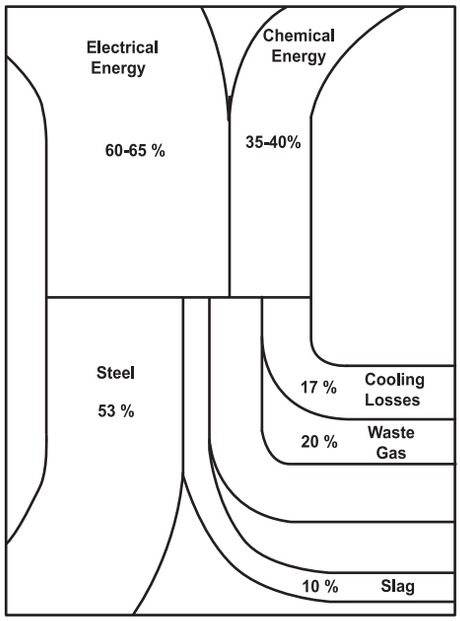

Figure 12.9 Sankey diagram for EAF

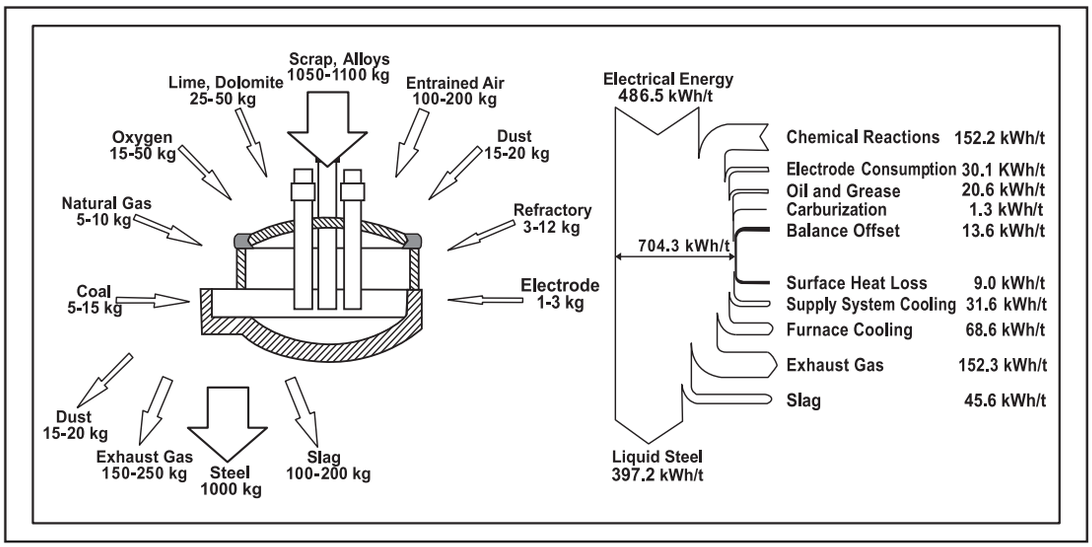

Figure 12.10 Energy and material balance of EAF

12.5 Case Example : Heat Balance of Rotary Hearth Furnace in a Secondary Steel Plant

In a rolling mill of a steel plant, slabs are reheated in a rotary hearth furnace (figure 12.11) for downstream rolling. The trial data is given below. Construct a heat balance.

Trial data

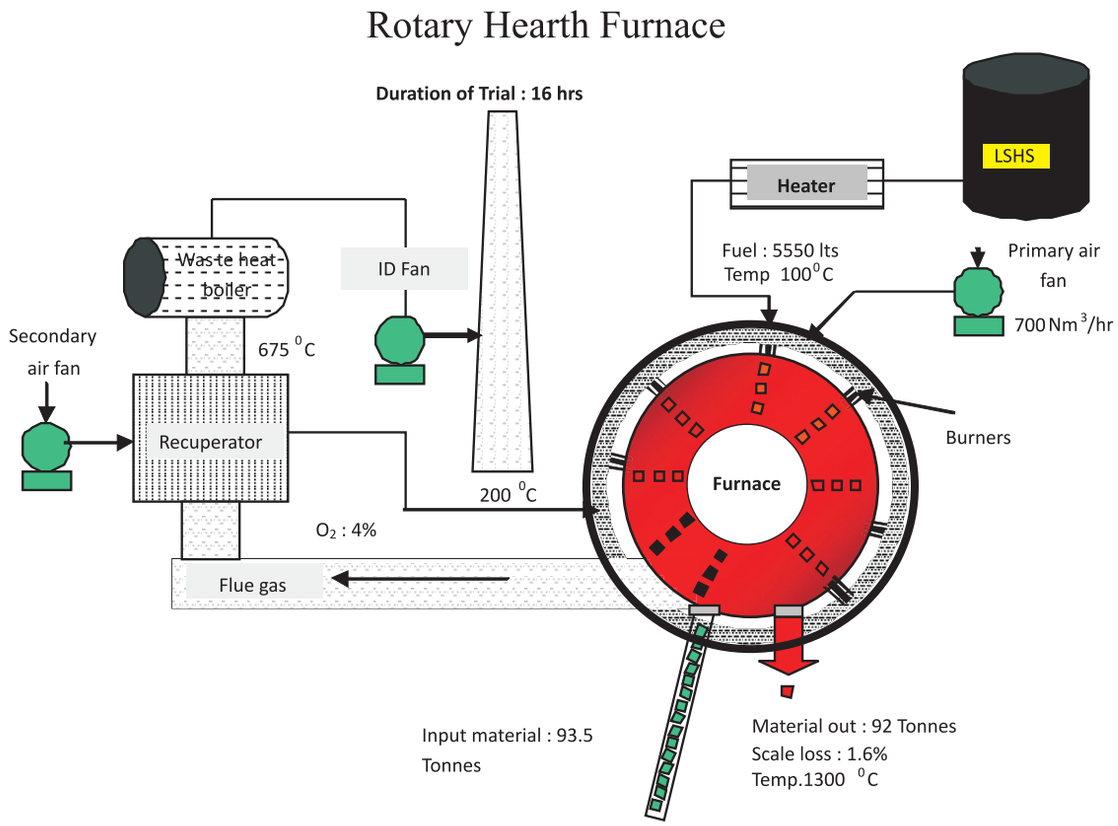

Figure 12.11 Heat Balance of rotary hearth furnace

Duration of trial	- 16 hours of rolling
Hydrogen in fuel	- 12 %
Water content	- 0.5 %
Calorific value of LSHS	- 10,200 kcal/kg
Specific gravity of LSHS	- 0.95
Specific heat of fuel	- 0.5 kcal/kg ^{0}C
Fuel consumption	- 5550 lts
Theoretical air required for combustion	- 14 kg of air / kg of fuel
Temperature of fuel before firing	- 100 ^{0}C
Temperature of furnace	- 1300 ^{0}C

Input material	- 93.5 T
Production	- 92 T
Scale loss	- 1.6 %
Specific heat of steel at 30 0C	- 0.11 kcal/kg ^{0}C
Specific heat of steel at 1300 0C	- 0.158 kcal/kg ^{0}C
Specific heat of scale at 1300 0C	- 0.215 kcal/kg ^{0}C
Heat of formation of scale	- 1335 kcal/kg of Fe
Ambient temperature	- 30 ^{0}C
Humidity of air	- 0.034 kg/kg of dry air
Atomising air	- 700 Nm3/hr
Secondary air temperature	- 200 ^{0}C
Specific heat of air at 200 ^{0}C	- 0.24 kcal/kg ^{0}C
Oxygen in flue gas	- 4 %
Flue gas temperature after air preheater	- 675 ^{0}C
Specific heat of flue gas	- 0.26 kcal/kg ^{0}C
Specific heat of water vapour	- 0.48 kcal/kg ^{0}C
Cooling water flow rate	- 28 m^3/hr
Temperature difference	- 10 ^{0}C
Number of doors	- 2 no's
Factor for radiation, φ	- 0.7 for rectangular door
Area of opening, A	- 1 m x 1m = 1 m2
Diameter of furnace	- 10 m
Height of furnace	- 2.5 m
Wall Surface area	- 3.14 x 10 x 2.5 - 78.5 m^2
Roof surface area	- 3.14 x 5^2 - 78.5 m^2
Average wall temperature	- 65 ^{0}C
Average roof temperature	- 100 ^{0}C

Solution:

Calculation of product rate

Production rate	- 92 / 16 - 5.75 T/hr

Calculation of fuel rate

Fuel rate	- 5550 / 16 - 347 liters/hr

Calculation of specific fuel consumption

Specific fuel consumption
- 347/5.75
- 60 lts/T of billet
- 60 x 0.95
- 57 kg/T of billet

Calculation of specific atomizing air

Atomizing air flow rate
- 700 Nm^3/hr
- 700 x 1.28
- 896 kg/hr

Specific atomizing air rate
- 896/5.75
- 156 kg of air /T of billet

Calculation of specific secondary air

Theoretical air required for combustion (from stoichiometric equation)
- 14 kg of air / kg of fuel

Excess air
- $\frac{4}{(21-4)} \times 100$
- 24%

Total air supplied to furnace
- 57 x {14 x (1+0.24)}
- 991 kg of air /T of billet

Secondary air required for combustion
- 991 - 156
- 835 kg/T of billet

Calculation of specific flue gas

Flue gas quantity
- 991 + 57
- 1048 kg/T of billet

Calculation of specific water vapour

Water vapour carried over in flue gas
- ($9H_2$ + water content)/100
- ((9 x 12) + 0.5)/100
- 1.085 kg/kg of fuel
- 1.085 x 57
- 62 kg/T of billet

Calculation of specific scale formation

Percentage scale formation
- 1.6 %
- $\frac{5.75 \times 0.016 \times 1000}{5.75}$
- 16 kg/T of billet

Calculation of specific cooling water

Cooling water flow rate	- 28 m^3/hr
Specific cooling water	- 28000/5.75 - 4869.6 kg/T of billet

Calculation of Heat Input

a) Combustion heat of fuel	- 57 x 10200 - 5,84,560 kcal/ T of billet
b) Sensible heat of fuel	- 57 x 0.5 x (100 – 30) - 2006 kcal/T of billet
c) Sensible heat of atomizing air	- 156 x 0.23 x (30 – 30) - 0 kcal/T of billet
d) Quantity of heat brought in by charged steel	- 1016 x 0.11 x (30 – 30) - 0 kcal/T of billet
e) Heat of formation of scale	- 16 x 1335 - 21,360 kcal/T of billet
Total heat input	- a + b + c + d + e + f - 5,84,560+ 2,006+ 0 + 0 + 21,360 **- 6,07,926 kcal/T of billet**

Calculation of Heat output

a) Sensible heat of steel	- 1000 x 0.158 x (1300 – 30) - 2,00,660 kcal/T of billet
b) Sensible heat of scale	- 16 x 0.215 x (1300 – 30) - 4,369 kcal/T of billet
c) Sensible heat of dry flue gas	- (1048– 62) x 0.26 x (675 – 30) - 1,65,389 kcal/T of billet
d) Heat loss due to formation of water vapour from fuel	- 62 x {584 + 0.48 x (675– 30)} - 55,564 kcal/T of billet
e) Heat loss due to formation of water vapour from air	- 991 x 0.034 x 0.48 x (675 – 30) - 10,433 kcal/T of billet
f) Heat taken away by cooling water	- 4869.6 x 10 - 48696 kcal/T of billet

g) Calculation of radiation heat losses

i) Radiation heat loss through - *hours x A x φ x 4.88 [(T$_f$/100)^4 – (T$_0$/100)^4]/T of billet* furnace opening

- 1 x 1 x 0.7 x 4.88 {(1573/100)^4 – (303/100)^4}/5.75
- 36,322 x 2 kcal/T of billet
- 72,643 kcal/T of billet

ii) Surface radiation losses

Q - a x $(t_1-t_2)^{1.25}$ + 4.88 x E x {$((t_1 + 273)/100)^4$ - $((t_2 + 273)/100)^4$}

Radiation heat losses through walls

- 2.2 x $(65-30)^{1.25}$ + 4.88 x 0.8 x {$((65 + 273)/100)^4$ - $((30 + 273)/100)^4$}
- 187.3 + 4.88 x 0.8 x (130 – 84)
- 187.3 + 179.6
- 366.9 kcal/m^2 hr

Wall surface losses
- 78.5 x 366.9
- 28801 kcal/hr
- 28801/5.75
- 5009 kcal/T of billet

Radiation heat losses through roof

- 2.8 x $(100-30)^{1.25}$ + 4.88 x 0.8 x {$((100 + 273)/100)^4$ - $((30 + 273)/100)^4$
- 567 + 4.88 x 0.8 x (193.6 – 84)
- 567 + 428
- 995 kcal/m^2 hr

Roof surface losses
- 78.5 x 995
- 78108 kcal/hr
- 78108 /5.75
- 13584 kcal/T of billet

Total radiation loss from furnace – 5,009 + 13,584 + 72,643
- 91,236 kcal/t of billet

Total heat output

- a + b + c + d + e + f + g
- 2,00,660 + 4,369 + 1,65,389 + 55,564 + 10,433+ 48696 + 91,236
- 5,76,347 kcal/T of billet

Heat balance table

Heat Input			Heat output		
Item	**kcal/T**	**%**	**Item**	**kcal/T**	**%**
Combustion heat of fuel	5,84,560	96.16	Sensible heat of steel	2,00,660	33.00
Sensible heat of fuel	2006	0.33	Sensible heat of scale	4,369	0.72
Sensible heat of atomizing air	0	0.00	Sensible heat of dry flue gas	1,65,389	27.2
Quantity of heat brought in by charged steel	0	0.00	Heat loss due to formation of water vapour from fuel	55,564	9.1
Heat of formation of scale	21,360	3.55	Heat loss duet to formation of water vapour from air	10,433	1.7
			Heat taken away by cooling water	48,696	8.0
			Radiation heat losses	91,236	15.0
			Others	31,579	5.2
Total	**6,07,926**	**100**		**6,07,926**	**100**

Solved Example:

In a steel plant, daily sponge iron production is 500 tons. The sponge iron is further processed in a steel melting shop for production of ingots. The yield from converting sponge iron into ingots is 88%. The plant has a coal fired captive power station to meet the entire power demand of the steel plant.

The base year (2011) and current year (2012) energy consumption data are given below:

Parameters	Base Year (2011)	Current Year (2012)
Sponge iron production	500 T/day	500 T/day
Specific coal consumption for sponge iron production	1.2 T/ T of Sponge Iron	1.1 T/T of Sponge Iron
Specific power consumption for sponge iron production	120 kWh/ T of Sponge Iron	100 kWh/ T of Sponge Iron
Yield ,in converting sponge iron into ingot in steel melting shop	88%	88%

Specific power consumption in steel melting shop to produce ingots	950 kWh / T of Ingot	900 kWh / T of Ingot
Captive power station heat rate	3500 kcal/ kWh	3200 kcal / kWh
GCV of coal	5000 kcal /kg	5000 kcal /kg

i) Calculate the specific energy consumption of the plant in Million kcals / Ton of finished product (Ingot) for the base year as well as for the current year

ii) Reduction in Coal consumption per day in current year compared to base year for the plant

Solution:

i) Specific energy consumption of the plant For Base Year

Specific energy consumption for sponge iron	= 1200 kgx 5000 + 120 kWhx 3500 = 6.42 million kcal/ Ton of SI
Total energy consumption for sponge iron /day	6.42 X 500 = 3210 million kcal
Actual production considering 88% yield from sponge iron to ingot conversion	= 500 Tons x 0.88 = 440 Tons / day
Specific energy consumption for ingot	= 950 kWhx 3500 = 3.325 million kcal/ ton of ingot
Total energy consumption for ingot production per day	3.325X 440=1463 million kcal
Plant specific energy consumption for production of finished product (ingot) during base year	(3210 +1463) / 440 = 10.62 million kcal/ ton

Specific energy consumption of the plant For Current Year

Specific energy consumption for sponge iron	= 1100 kgx 5000 + 100 kWhx 3200 = 5.82 million kcal Ton of SI
Total energy consumption for sponge iron /day	5.82 X 500 = 2910 million kcal
Actual production considering 88% yield from sponge iron to ingot conversion	= 500 T X 0.88 = 440 Tons / day
Specific energy consumption for ingot	= 900Kwh x 3200 = 2.88 million kcal/ ton of ingot
Total energy consumption for ingot production per day	2.88 X 440 = 1267.2 million kcal
Plant specific energy consumption for production of finished product (ingot) during current year	= (2910+1267.2)/440 = 9.49 million kcal/ ton

ii) Reduction in coal consumption

Energy saving in sponge iron plant	= (6.42-5.82) x500	= 300 million kcals/day
Energy saving in steel melting plant	= (3.325-2.88) x 440	= 195.8 million kcal/day
Total energy saving	= 300 + 195.8	= 495.8 million kcal/day
Equivalent coal reduction (saving)	= 495.8 x 10^6 /5000	= 99.16 Tons per day

QUESTIONS	
S-1	What are the raw materials used in pig iron production?
S-2	Why BOF is preferred to OHF in steel pant?
S-3	What are the raw materials used in secondary steel manufacturing with EAF?
S-4	What are the functions of coke in blast furnace steel making?
S-5	What are the inputs raw materials used in sintering?
S-6	Using chemical equation how sulfur is removed from iron in a blast furnace?
S-7	How is blast furnace top gas utilized?
S-8	What is the range of temperatures encountered in a basic oxygen furnace?
S-9	Explain what do you understand by "Near Net Shape Casting"?
S-10	What are sources of heat input in an electric arc furnace?
L-1	An annealing furnace in a steel plant has the charging door of 1.5 m^2 area. The furnace temperature is 800 ^oC. The ambient temperature is 30 ^oC. If the factor of radiation is 0.75. Find out the heat loss per hour.
L-2	The actual composition of raw coke oven gas is as follows Water vapour - 47%, H_2 - 29%, CH_4 - 13%, N_2 - 5%, CO - 3%, CO_2 - 2%, Hydrocarbon – 1%. Find out the composition in dry basis.
N-1	The composition of blast furnace gas by volume is as follows CO – 27%, H_2 - 2%, CO_2 – 11%, N_2 - 60%. i) Calculate the stoichiometric air for combustion ii) Calculate the gross calorific value of gas in kcal/Nm^3 iii) Calculate the net calorific value of gas in kcal/Nm^3 iv) If 25,000 Nm^3/hr of gas is available and is to be co-fired in a coal fired boiler. How much coal it can replace if the GCV of coal is 4300 kcal/kg.

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Chapter 13

Performance Assessment of Cement Industry

Learning Objectives

In this chapter you will learn about

- Cement manufacturing process
- Energy interaction in cement manufacture
- Material and energy balance calculations for cement kiln
- Heat balance of raw mill

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Understand the process of cement making**
- ✓ **Perform material and energy balance of cement kiln**
- ✓ **Estimate specific energy consumption**
- ✓ **Perform material and energy balance of raw mill**

13. ENERGY PERFORMANCE ASSESSMENT OF CEMENT INDUSTRY

13.1 Introduction

Cement acts as a bonding agent, holding particles of aggregate together to form concrete. Cement production is highly energy intensive and involves the chemical combination of calcium carbonate (limestone), silica, alumina, iron ore, and small amounts of other materials. Cement is produced by burning limestone to make clinker, and the clinker is blended with additives and then finely ground to produce different cement types. Desired physical and chemical properties of cement can be obtained by changing the percentages of basic chemical components (CaO, Al_2O_3, Fe_2O_3, MgO, SiO_2, etc.). Mostly the cement produced is of Portland cement. Other cement types include white, masonry, slag, aluminous, and regulated-set cement. Cement production involves quarrying and preparing the raw materials, producing clinker through pyroprocessing the materials in huge rotary kilns at high temperatures, and grinding the resulting product into fine powder.

13.2 Cement Manufacturing Process

The basic process of Cement production as shown in Figure 13.1 involves

- Acquisition of raw materials
- Preparation of the raw materials for pyroprocessing
- Pyroprocessing of the raw materials to form Portland cement clinker, and,
- Grinding the clinker to Portland Cement

Description of production processes

Mining: Limestone, the key raw material is mined in the quarries with compressed air drilling and subsequently blasting with explosives. The mined limestone is transported through dumpers or ropeways to the plant. Surface mining is gradually gaining ground because of its eco friendliness.

Crushing: The limestone as mined is fed to a primary and secondary crusher where the size is reduced to 25 mm. Of late even a tertiary crusher is used to further reduce the inlet size to the mill. The crushed limestone is stored in the stockpile through stacker conveyors. The crushed limestone, bauxite and ferrite are stored in feed hoppers from where they are fed to the raw mill via weigh feeders in the required proportion.

Raw Materials Preparation: For dry-process cement making, the raw materials need to be ground into a flowable powder before entering the kiln. Generally ball mills or vertical roller mills are used. Modern cement plants mostly use vertical roller mills. Roller mills for grinding raw materials and separators or classifiers for separating ground particles are the two key energy consuming pieces of

equipment at this process stage. Along with grinding simultaneous drying of raw materials using hot gases from the preheater exhaust also takes place.

Coal Milling: In plants using coal, coal mills are part of the system to provide dried pulverized coal to kiln and precalciner. The raw coal from stock yard is crushed in a hammer crusher and fed to the coal mill. The coal mill can be an air swept ball mill or vertical roller mill where the coal particles are collected in the bag filter through a grit separator. The required size is 80 % on 90 micron and less than 2% on 212 micron. Hot air generated in a coal fired furnace or hot air from clinker cooler/preheater exhaust is used in the drying of coal in the mill.

Pyro processing: The function of the kiln in the cement industry is to first convert $CaCO_3$ into CaO and then react Silica, Aluminum Oxide, Ferric Oxide, and Calcium Oxide with the free lime to form clinker compounds: C3S, C2S, C3A, and C4AF. The raw material mix enters the kiln at the elevated end, and the combustion fuels generally are introduced into the lower end of the kiln in a countercurrent manner. The materials are continuously and slowly moved to the lower end by rotation of the kiln. Pulverized coal, gas, pet coke or Oil are the fuels generally used. This system transforms the raw mix into clinkers, which are gray, glass-hard, spherically shaped nodules that range from 0.32 to 5.1 centimeters (cm) in diameter. The chemical reactions and physical processes that constitute the transformation are quite complex, but they can be viewed conceptually as the following sequential events:

- Evaporation of uncombined water from raw materials as material temperature increases to 100 ^oC
- Dehydration as the material temperature increases from 100 ^oC to approximately 430 ^oC to form oxides of silicon, aluminum, and iron;
- Calcination, during which carbon dioxide (CO_2) is evolved, between 900 ^oC and 982 ^oC to form CaO
- Reaction of the oxides in the burning zone of the rotary kiln to form cement clinker at temperatures of approximately 1510 ^oC

Pre heater and Pre calciner: Preheaters are cyclones which are arranged vertically, in series, and are supported by a structure known as the preheater tower. Hot exhaust gases from the rotary kiln pass counter currently through the downward-moving raw materials in the preheater vessels. Compared with the simple rotary kiln, the heat transfer rate is significantly increased, the degree of heat utilization is more complete, and the process time is markedly reduced owing to the intimate contact of the solid particles with the hot gases. The improved heat transfer allows the length of the rotary kiln to be reduced or in other words for the existing kiln if retrofitted, increases the production.

Additional thermal efficiencies and productivity gains have been achieved by diverting some fuel to a calciner vessel at the base of the preheater tower. This system is called the preheater/precalciner process. While a substantial amount of fuel is used in the precalciner, at least 40 percent of the thermal energy is required in the rotary kiln.

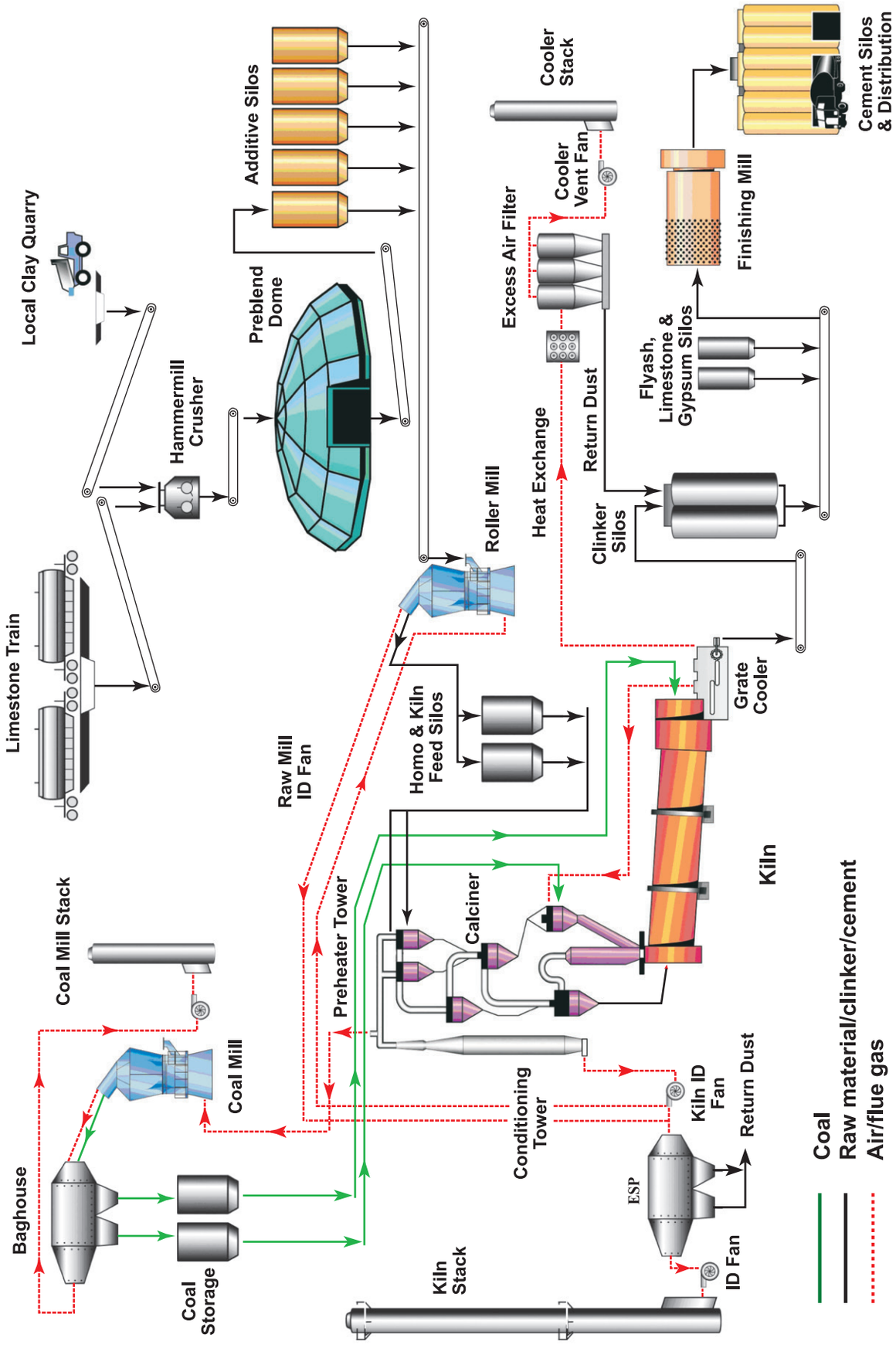

Figure 13.1. Typical flow diagram of cement manufacturing plant

Upto 95 % of the rawmeal gets calcined before entering the kiln. Calciner systems sometimes use lower-quality fuels (e.g., less-volatile matter) as a means of improving process economics.

From pre-heater and pre-calciner, 60 % of flue gases travel towards raw mill and 40 % to conditioning tower where water injection is used to condition the gases. These gases are ultimately passed through electrostatic precipitator (ESP) for the maximum removal of particulate matters.

Clinker Cooler: The hot output from the kiln (clinker) is cooled from 1450 ^{o}C to 130 ^{o}C in the grate cooler with a series of fans. The cooler has two tasks: to recover as much heat as possible from hot clinker so as to return it to the process; and to reduce the clinker temperature to a level suitable for the equipment downstream. The hot air from recuperation zone is used for main burning air (secondary air) and precalciner fuel (tertiary air). The remaining air is sent to the stack through multiclones or ESP. Once clinker leaves the kiln it must be cooled rapidly to ensure the maximum yield for the compound that contributes to the hardening properties of cement. The main cooling technologies are the reciprocating grate cooler and the tube or planetary cooler.

Finish Milling: In this final process step, the cooled clinker is mixed with additives to make cement and ground using the mill technologies described earlier. The grinding process occurs in a closed system with an air separator that divides the cement particles according to size. Material that has not been completely ground is sent through the system again. Finish milling is the grinding of clinker to produce a fine grey powder. Gypsum ($CaSO_4$) is blended with the ground clinker, along with other materials, to produce finished cement. Gypsum controls the rate of hydration of the cement in the cement-setting process. The cement thus produced is collected in the bagfilter and taken to cement silos through a vertical pneumatic pump. The energy used for cement grinding depends on the type of materials added to the clinker and on the desired fineness of the final product. Cement fineness is generally measured in a unit called Blaine, which has the dimensions of cm^2 /g and gives the total surface area of material per gram of cement. Higher Blaine indicates more finely ground cement, which requires more energy to produce. Portland cement commonly has a Blaine of 3000-3500 cm^2 /g.

Energy flow

The cement making process is highly energy intensive accounting for nearly 40 – 50 % of the production costs. This provides ample opportunities for reducing energy consumption as many of the cement plants in developing countries consume much more than the best achieved figures in developed countries.

Electrical Energy:

The energy flows in a typical cement plant is given in the Figure 13.2 below. The major electrical energy consumption areas are mill drives, fans and conveying systems.

About 30% of electric power is consumed for finish grinding, and below 30% is consumed by the clinker burning process. Raw mill circuit is another major consumer accounting for 24 % of the energy.

The raw mill circuit and finish grinding process mainly consumes electric power for the mill, and the clinker burning process mainly for the fan.

Thermal Energy:

Thermal energy accounts for almost half the energy costs incurred in cement manufacture. A variety of fuels such as coal, pet coke, gas and oil in addition to unconventional fuels such as used tires, incinerable hazardous wastes, agro residues etc are used in the cement plant. The major use of thermal energy is in the kiln and precalciner. In plants using coal, an external coal or oil fired furnace is used for generation of hot air required for coal mills. The average thermal energy consumption kcal/kg of Clinker is given in the table 13.1.

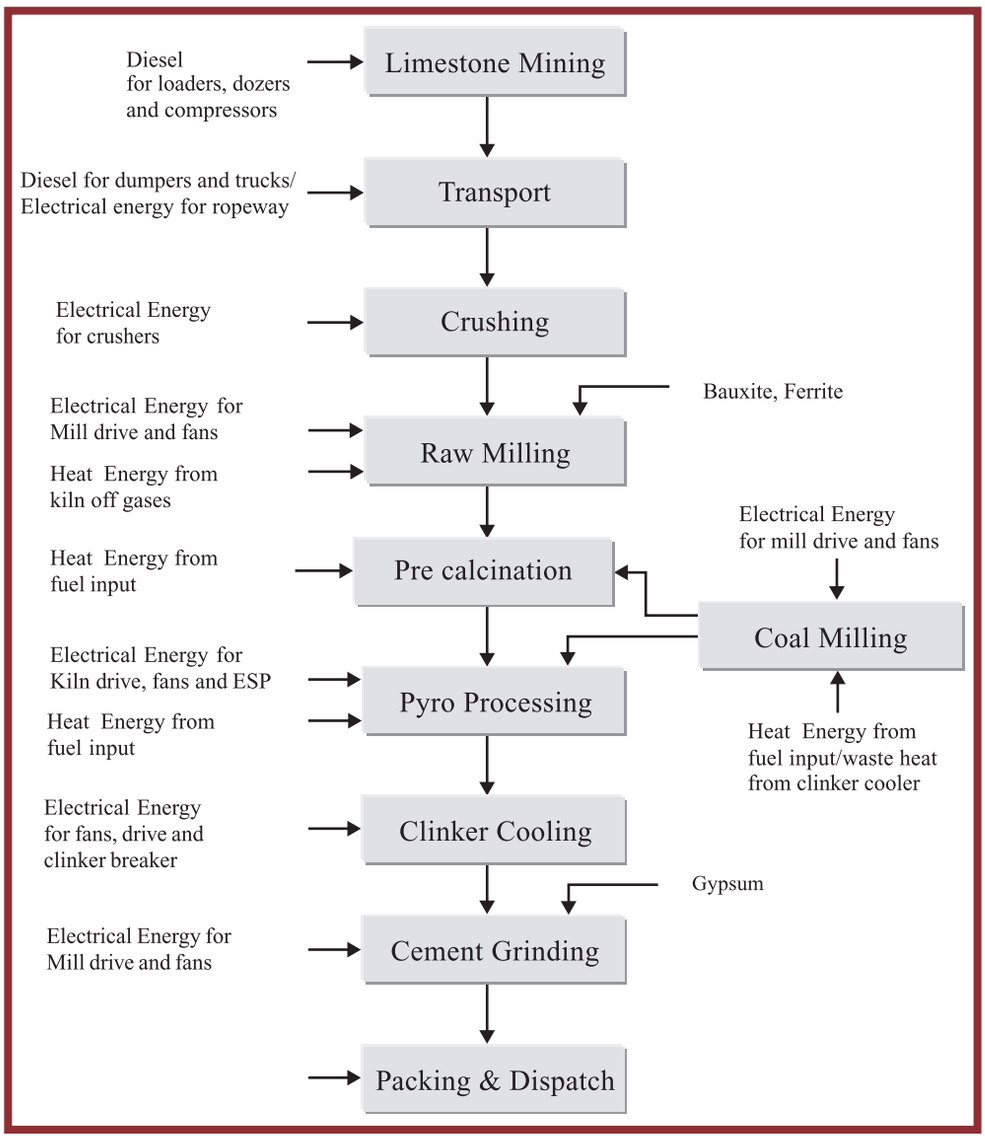

Figure 13.2. Process flow diagram and type of energy used in cement manufacturing

Table 13.1. Average and Best Practice Energy Consumption Values for Indian Cement Plants by Process.

Process	Unit	India Average	World Best Practice
Raw Materials Preparation			
Coal mill	kWh/t clinker	8	2.4
Crushing	kWh/t clinker	2	1.0
Raw mill	kWh/t clinker	28	27
Clinker Production			
Kiln & cooler	kcal/kg of clinker	770	680
Kiln & cooler	kWh/t clinker	28	22
Finish Grinding			
Cement mill	kWh/t cement	30	25
Miscellaneous			
Utilities: mining & transportation	kWh/t clinker	1.6	1.5
Utilities: packing house	kWh/t cement	1.9	1.5
Utilities: misc.	kWh/t cement	2.0	1.5
Total Electric	kWh/t cement	95*	77

* Note : 1. The specific energy consumption of cement manufacture depends upon several factors such as raw meal quality (grindability index of limestone), fuel quality, pre-heating stages, percentage of blending of fly ash, slag and other mineral matter with clinker, fineness of the final product (blain number), plant vintage, capacity and technology employed etc.

2. The best reported values by Indian cement plants were 66.2 kWh/t cement and 687 kcal/kg clinker in 2008-09 (source: National Energy Conservation Award - 2009)

13.3 Material and Energy balance

The cement process involves gas, liquid and solid flows with heat and mass transfer, combustion of fuel, reactions of clinker compounds and undesired chemical reactions that include sulphur, chlorine, and Alkalies. It is important to understand these processes to optimize the operation of the cement kiln, diagnose operational problems, increase production, improve energy consumption, lower emissions, and increase refractory life. A heat balance should be constructed for the preheater, kiln, cooler and the heat output values should be compared with standard values.

Example 13.1. Heat Balance (Indirect Method) of a Six-Stage Preheater Inline Calciner (ILC) Kiln

INPUT DATA

Process measurements in kiln system

Location	Temp (^{0}C)	Static Pressure (mmWG)	$(\Delta P_{rms})_{avg}$ (mmWG)	Oxygen (% v/v)	CO (% v/v)	CO_2 (% v/v)	Duct Area (m^2)
Preheater outlet	316	-655	16.39	5.1	0.1	24.7	8.299
Calciner outlet	882	- 110	-	1.0	0.15	-	-
Cooler vent	319	- 35	16.2	-	-	-	6.996

Note – Gas analysis measurement on dry basis

Moisture in kiln feed = 0.6%	Clinker temperature (t_c) = 130 ^{o}C
Moisture in fine coal = 1.0%	Kiln feed temperature(t_{kf}) = 73 ^{o}C
Kiln feed rate = 284.98 TPH	Kiln/PC fine coal temperature(t_{ca}) = 68 ^{o}C
Kiln out put rate (Clinker) = 171.68 TPH	Primary air temperature = 48 ^{o}C
Return dust in preheater @ 8 %of kiln feed rate	Ambient temperature(t_A) = 33 ^{o}C
Kiln feed to Clinker factor (gross) = 1.66	Reference temperature (t_r) = 20 ^{o}C
Kiln feed to Clinker factor (net) = 1.527	NCV of fine coal = 6084 kcal/kg coal

Chemical analysis of Clinker (Loss free basis)

Constituents Determined	%
SiO2	21.40
Fe2O3	5.26
Al2O3	5.62
CaO	64.96
MgO	1.18

m_{d-} mass of preheater return dust in pre-heater exit gases, kg/kg Clinker
$m_{e_}$ mass of pre-heater exit gases, kg/kg Clinker
m_c- mass of Clinker, kg
m_{ce}- mass of cooler exhaust gases, kg/kg Clinker
m_{mr}- mass of moisture in kiln feed, kg/kg Clinker
m_{mc}- mass of moisture in coal, kg/kg Clinker
m_{ic}- mass of carbon monoxide in preheater exit gases, kg/kg Clinker
m_{kf} -mass of kiln feed as fed to the kiln system, kg/kg Clinker
m_{co}- mass of cooling air from the cooler fans, kg/kg Clinker
m_P- mass of total primary air, kg/kg Clinker
m_{fuel}- mass of fuel consumed / fed to the kiln system, kg/kg Clinker
m_{lk} – mass of leaking air, kg/kg Clinker

Cp_d- mean specific heat of preheater return dust, kcal/kg ^oC
Cp_e- mean specific heat of pre-heater exit gases. kcal/kg ^oC
Cp_c- mean specific heat of Clinker. kcal/kg ^oC
Cp_{ce}- mean specific heat of cooler exhaust air. kcal/kg ^oC
Cp_s- mean specifc heat of water vapour. kcal/kg ^oC
Cp_{kf}- mean specific heat of kiln feed, kcal/kg ^oC
Cp_{air}- mean specific heat of cooling air, kcal/kg ^oC
P_t- pitot tube coefficient (0.844)
P_s- static pressure, mmWG
$(\Delta P_{rms})_{avg}$ – average of root mean square of dynamic/velocity pressure , mmWG
g- acceleration due to gravity (9.81 m/sec^2)
t_e- pre-heater exit gas temperature, ^oC
t_c-Clinker temperature, ^oC
t_{ce}- cooler exhaust air temperature, ^oC
t_{kf}-kiln feed temperature, ^oC
t_{co}-temperature of cooling air, ^oC
t_p-temperature of primary air, ^oC
T_A – ambient temperature, K
MW- molecular weight

ρ_{stp} - density at standard temperature and pressure

$\rho_{t,p}$ - density at prevailing temperature and pressure

L- latent heat of steam (540 kcal/kg)
σ - Stefan-Boltzmann constant
ε - emissivity or emittance
Cl – Clinker
ΔH_R – heat of formation of Clinker

Basis : 1 kg Clinker

HEAT OUTPUT

a) <u>Heat of formation of Clinker</u>

$\Delta H_R = 2.22Al_2O_3+6.48MgO+7.646CaO–5.116SiO_2–0.59Fe_2O_3$
$= (2.22 \times 5.62)+(6.48 \times 1.18)+(7.646 \times 64.96)–(5.116 \times 21.4)–(0.59 \times 5.26)$
$= 404.22$ kcal/kg Clinker

b) <u>Heat in pre heater exit dust:</u>

$Q_l = m_d \times Cp_d \times (t_e - t_r)$
Where, m_d = kiln feed to Clinker factor × preheater return dust
$= 1.66 \times 0.08$
$= 0.1328$ kg/kgCl
$Q_l = 0.1328 \times 0.23 \times (316-20)$ (Cp_d= 0.23 kcal/kg ^0C)
$= 9.04$ kcal/kg Clinker

c) Heat in preheater exit gases:

$Q_2 = m_e \times Cp_e \times (t_e - t_r)$

Where, m_e = Mass of preheater exit gas in kg per kg of Clinker (on dry basis)

Cpe = specific heat of preheater exit gas in kcal/kg ^oC

t_e = temperature of exit gas in ^oC

t_r = reference temperature in ^oC

As moisture from kiln feed and fine coal in preheater exit gases is very small, density of gas is estimated on dry basis.

Now, density of gases at preheater exit :

$$\rho_{stp} = \frac{(O_2\% \text{ x MW}) + (CO_2\% \text{ x MW}) + ((N_2 + CO)\% \text{ x MW})}{22.4 \text{ x } 100} \text{ kg / Nm}^3$$

$$\rho_{stp} = \frac{(5.1 \text{ x } 32) + (24.7 \text{ x } 44) + (70.2 \text{ x } 28)}{22.4 \text{ x } 100} \text{ kg / Nm}^3$$

$= 1.436 \text{ kg/Nm}^3$

$$\rho_{t,p} = \rho_{stp} \times \frac{10334^* - P_S}{10334} \times \frac{273}{273 + t_e} \text{ kg / m}^3$$

(* as per the altitude of the plant above MSL)

$$\rho_{t,p} = 1.436 \times \frac{10334 - 655}{10334} \times \frac{273}{273 + 316} \text{ kg / m}^3$$

$= 0.623 \text{ kg/m}^3$

Volume of pH exit gases (V^g) = Velocity ×Area of duct

Velocity $= P_t \times \sqrt{(2g(\Delta P_{rms})_{avg} / \rho_{t,p})}$ m/sec

$$= 0.844 \times \frac{\sqrt{2 \times 9.81 \times 16.39}}{\sqrt{0.623}}$$

$= 19.17$ m/sec

$V^g = 19.17 \text{ x } 8.299 \text{ m}^3/\text{sec}$

$= 159.1 \text{ m}^3/\text{sec}$

$= 159.1\text{x } 3600 \text{ x } (0.623/1.436) \text{ Nm}^3/\text{h}$

$= 248565 \text{ Nm}^3/\text{h}$

Specific Volume of preheater exit gases = V^g / kiln out put rate

$= 248565/171680 \text{ Nm}^3/\text{kg Clinker}$

$= 1.45 \text{ Nm}^3/\text{kg Clinker}$

$Q_2 = 1.45 \text{ x } 1.436 \text{ x } 0.247\text{x } (316 - 20)$, ($Cp_e = 0.247$ kcal/kg ^0C)

$= 152.23$ kcal/kg Clinker

d) Heat in Clinker from cooler discharge:

$Q_3 = m_c \times Cp_c \times (t_c - t_r)$

$= 1 \times 0.193 \times (130 - 20)$, ($Cp_c$ = 0.193 kcal/kg ^{0}C)

$= 21.23$ kcal/kg Clinker

e) Heat in cooler exhaust air:

$Q_4 = m_{ce} \times Cp_{ce} \times (t_{ce} - t_r)$

Where m_{ce} = mass of air, kg/kg clinker

Now, density of air at cooler exhaust

$$\rho_{t,p} = 1.293 \times \frac{10334^{*} - P_S}{10334} \times \frac{273}{273 + t_{ce}} \text{ kg/ m}^3$$

(ρ_{stp} =1.293 kg /Nm^3)

($*$ as per the altitude of the plant above MSL)

$$\rho_{t,p} = 1.293 \times \frac{10334 - 35}{10334} \times \frac{273}{273 + 319} \text{ kg/ m}^3$$

$= 0.594$ kg/ m^3

Volume of air at cooler exhaust (V^a) = velocity × Area of duct at which measurement carried out

Now, velocity $= P_t \times \sqrt{(2g(\Delta P_{rms})_{avg} / \rho_{t,p})}$ m / sec

$$= 0.844 \times \frac{\sqrt{2 \times 9.81 \times 16.2}}{\sqrt{0.594}} \text{ m / sec}$$

$= 19.52$ m/sec

$V^a = 19.52 \times 6.996$ m^3/sec

$= 136.56$ m^3/sec

$= 136.56 \times 3600 \times (0.594/1.293)$ Nm^3/h

$= 225849.96$ Nm^3/h

Sp. volume of cooler exhaust air = V^a / kiln out put rate

= 225849.96 /171680 Nm^3/kg Clinker

= 1.32 Nm^3/kg Clinker

$Q_4 = 1.32 \times 1.293 \times 0.244 \times (319 - 20)$ (Cp_{ce} = 0.244 kcal/kg ^{0}C)

$= 124.52$ kcal/kg Clinker

f) Heat of evaporation of kiln feed moisture:

$Q_5 = m_{mr} \times [(100 - t_{kf}) + L + Cp_s (t_e - 100)]$

Where, m_{mr} = kiln feed to Clinker factor × moisture in kiln feed

$= 1.66 \times 0.006$

$= 0.00996$ kg/kgCl

$Q_5 = 0.00996 \times [(100 - 73) + 540 + 0.457 \times (316 - 100)]$ (Cp_s = 0.457 kcal/kg ^{0}C)

$= 6.63$ kcal/kg Clinker

g) Heat of evaporation of fine coal moisture:

$Q_6 = m_{mc} \times [(100 - t_{ca}) + L + Cp_s (t_e - 100)]$

Where, m_{mc} = coal consumption in kg per kg Clinker ×moisture in fine coal

$Q_6 = m_{fuel} \times 0.01$ x [32 + 540 + 0.457 (316 – 100)]
= 6.707 x m_{fuel} kcal/kg Clinker

h) Heat loss due to incomplete combustion:

$Q_7 = m_{ic} \times 67636$

Where, Heat of combustion of CO = 67636 kcal/kg mole,

$$m_{ic} = \frac{CO\%}{100 \times 22.4} \times Specific Volume of\ preheater\ exit\ gases$$

$Q_7 = 6.47 \times 10^{-5} \times 67636$
= 4.38 kcal/kg Clinker

i) Radiation & Convection losses:

Radiation losses = $\sigma \times \varepsilon \times (T_s^4 - T_A^4)$ x Surface Area, kcal/h

σ = 4.88×10^{-8} kcal/$m^2 - k^4$

ε = 0.80 for oxidized steel

Convection Losses : (i) (if wind velocity $\leq$ 3m/sec)

$$= 80.33 \times \left(\frac{T_A + T_S}{2}\right)^{-0.724} \times (T_S - T_A)^{1.33} \times \text{Surface Area kcal/h,}$$

(ii) if (wind velocity $\geq$ 3m/sec)

$$(5.86 + 3.04 \times wind\ velocity) \times (T_S - T_A) \times Surface\ Area \text{ kcal/h,}$$

Sample calculation for radiation and convection losses from kiln:

Distance from kiln outlet (m)	Surface area (m^2)	Surface temp, T_s (K)	Radiation loss (kcal/h) (a)	Convection loss (kcal/h) (b)	Heat loss (kcal/h) (a+b)
1	12.41	566.00	45473.96	20268.51	65742.47
2	12.41	568.00	46180.59	20442.62	66623.21

Note: (1) Average measurements of surface temperature to be taken at 1 m interval along the length of kiln as well as tertiary air duct and calculations for radiation and convectional losses to be worked out as shown above.

(2) Here, wind velocity of 1.5 m/sec considered

Sample calculation for radiation and convection losses from preheater:

	Surface area (m^2)	Surface temp, T_s (K)	Radiation loss (kcal/h) (a)	Convection loss (kcal/h) (b)	Heat loss (kcal/h) (a+b)
Cyclone stage 1A					
Cylindrical portion	416	360	130386.88	101624.64	232011.52
Conical portion	106.6	386	55858.40	42772.84	98631.24
Cyclone top	16.62	364	5701.65	4446.51	10148.16
Meal pipe	20.93	429	20512.23	14264.0	34776.23
Duct from previous stage	250.33	369	95500.89	74378.05	169878.94

Note: (1) Average measurements of surface temperature to be taken on the surfaces of each cyclone of preheater and calculations for radiation and convectional losses to be worked out as shown above
(2) Here, wind velocity of 1.5 m/sec considered

Radiation and convection losses from kiln = 4013284.69 kcal/h
= 23.38 kcal/kg Clinker
Radiation and convection losses from tertiary air duct = 662684 kcal/h
= 3.86 kcal/kg Clinker
Radiation and convection losses from preheater = 3071821.42 kcal/h
= 17.89 kcal/kg Clinker
Radiation and convection losses from cooler assumed to be 3 kcal/kg Clinker.

Q_8 = Total radiation and convection losses = 23.38+3.86+17.89+3 kcal/kg Clinker
= 48.13 kcal/kg Clinker

HEAT INPUT

j) Sensible heat in kiln feed

$Q_9 = m_{kf} \times Cp_{kf} \times (t_{kf}-t_r)$
Where, m_{kf} is kiln feed to Clinker factor
Q_9 = 1.66 x 0.213 x (73 – 20) (Cp_{kf} = 0.213 kcal/kg ^{0}C)
=18.74 kcal/kg Clinker

k) Sensible heat in cooling air

$Q_{10} = m_{co} \times Cp_{air} \times (t_A-t_r)$
Where, m_{co} = Mass of cooling air, kg/kg clinker
Standard density of air =1.293 kg/Nm^3
P_s - Static Pressure, mmWG = -35

$$\rho_{t,p} = 1.293 \times \frac{10334 - P_S}{10334} \times \frac{273}{273 + t_A} \text{ kg / m}^3$$

= 1.15 kg/m^3

Cooling air fans flow

Fan	Velocity (m/sec)	Area (m^2)	Volume of air (m^3/sec)	P_S (mmWG)	Volume of air (Nm^3/sec)
FN 1	21.29	0.6364	13.55	-33	12.05
FN 2	14.05	0.6364	8.94	-28	7.95
FN 3	14.47	0.6364	9.21	-35	8.19
FN 4	12.25	0.5029	8.861	-46	7.87
FN 5	16.93	0.5029	8.541	-48	7.58
FN 6	17.62	0.5029	8.61	-30	7.66
FN 7	14.40	0.5029	7.24	-29	6.44
FN 8	23.20	0.7857	11.67	-12	10.40
FN 9	21.65	0.6364	13.77	-36	12.24
FN 10	27.78	0.6364	17.68	-31	15.73
Total					**96.11**

Sp. volume of cooling air = 96.11 Nm^3/sec
= 2.01 Nm^3/kg Clinker

Q_{10} = 2.01 x 1.293 x 0.237 x (33 – 20), (Cp_{air} = 0.237 kcal/kg ^{0}C)
= 8.0 kcal/kg Clinker

l) Sensible heat in primary air

Q_{11} = m_p x Cp_{air} x (t_p-t_r)

Where, m_p =Primary air to kiln + Primary air to calciner, to be calculated as above
= 0.073 + 0.009 kg/kg Clinker
= 0.082 kg/kg Clinker

$\therefore Q_{11}$ = 0.082 x 0.237 x (48 – 20)
= 0.54 kcal/kg Clinker

m) Sensible heat in leaking air

$Q_{12} = m_{lk}$ x Cp_{air} x (t_A-t_r)

$m_{lk} = m_e$ –[m_e/(1+leaking air)]

% Leaking air = [(O_2 % at PH O/L - O_2 % at PC O/L)/(21-O_2 % at PH O/L)]x 100

$$= \frac{5.1-1.0}{21-5.1} \times 100 = 26\%$$

$$\therefore m_{lk} = 1.45 - \frac{1.45}{1+0.26} = 0.299$$

Q_{12} = 0.299 x 1.293 x 0.237x (33-20)
= 1.19 kcal/kg Clinker

n) Sensible heat of fuel:

Q_{13} = m_{fuel} x Cp_{fuel} x (t_{fue} – t_r)
= m_{fuel} x 0.288 x (68-20), (Cp_{fuel} = 0.288 kcal/kg ^{0}C)
= 13.824m_{fuel} kcal/kg Clinker

0) Heat from coal combustion (Q_{14})

Q_{14} = m_{fuel} x NCV
= 6084 x m_{fuel}

Typical Heat Balance of Kiln

Item	kcal/kg Clinker
Heat Output	
Heat of formation of Clinker (ΔH_R)	404.22
Heat in dust in preheater exit gases (Q_1)	9.04
Heat in preheater exit gases (Q_2)	152.23
Heat in Clinker leaving cooler (Q_3)	21.23
Heat in cooler exhaust air (Q_4)	124.52
Heat for evaporation of moisture of kiln feed (Q_5)	6.63
Heat for evaporation of moisture of fine coal (Q_6)	6.707m_{fuel}

Heat in incomplete combustion (Q_7)	4.38
Radiation & convection losses (Q_8)	48.13
Total Heat output	**770.38 + 6.707m_{fuel}**
Heat Input	
Sensible heat in kiln feed (Q_9)	18.74
Sensible heat in cooling air (Q_{10})	8
Sensible heat in primary air (Q_{11})	0.54
Sensible heat in leaking air (Q_{12})	1.19
Sensible heat in fuel (Q_{13})	13.824m_{fuel}
Heat from coal combustion(Q_{14})	6084m_{fuel}
Total Heat Input	**28.47 + 6097.824m_{fuel}**

Based on the above

Total Heat Input = Total Heat Output

Hence, $28.47 + 6097.824m_{fuel} = 770.38 + 6.707m_{fuel}$

$\therefore m_{fuel}$ = 0.1218 kg/kg Clinker (12.18% Clinker basis)

Hence, heat from coal combustion (Q_{14}) = 6084 x 0.1218 = 741.05 kcal/kg Clinker

Total Heat Input = Total Heat Output

$$(Q_9 + Q_{10} + Q_{11} + Q_{12} + Q_{13} + Q_{14}) = (\Delta H_R + Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6 + Q_7 + Q_8)$$

{18.74 + 8 + 0.54 + 1.19 + (13.824 x 0.1218) + (6084 x 0.1218)}= (404.22 + 9.04 + 152.23 + 21.23 + 124.52 + 6.63 + (6.707 x 0.1218) + 4.38 + 48.13)

771.2 kcal/kg Clinker = 771.2 kcal/kg Clinker

Inference

Significant abnormal heat loss is almost invariably associated with preheater exhaust and/or with cooler exhaust. As can be seen from the typical heat balance given here, of a total heat consumption of 771 kcal/kg of Clinker, 404 kcal/kg of Clinker is for chemical reaction. Of the remaining 367 kcal/kg of Clinker which goes as heat losses, 75 % of it is due to losses through kiln and cooler stacks. The challenge is to reduce these exhaust gas heat losses.

High exhaust heat loss may be mitigated by:

1. Reduce losses due to CO formation
    - Good combustion in the main burner to avoid local reducing conditions in the presence of excess oxygen
    - Sufficient air/fuel mixing and retention time in the calciner for complete combustion
2. Reduce precalciner exhaust gas quantity
    - Maximize heat recuperation from the cooler
    - Minimize excess air without compromising combustion

- Avoid over-burning with consistence kiln feed chemistry and with constant feed and fuel rates (every 0.1 % free-lime below the optimum waste upto 14 kcal/kg Clinker)
- Minimize false air at kiln seals and preheater ports

3. Reduce preheater exhaust gas temperature
   - Good meal distribution in gas duct by design and maintenance of splash plates and splash boxes
   - High cyclone efficiency so that hot meal is not carried up the preheater
4. Reduce cooler exhaust
   - Maximum heat recuperation by control of air flow, Clinker distribution and Clinker granulometry.
   - Minimize false into firing hood and kiln discharge seal

13.4 Raw Mill

Raw milling, as one of the major cement process step, must produce sufficient kiln feed meeting targets for fineness, chemical composition and moisture to sustain required kiln production. The preheater gas at around 300 ^{o}C is used in the mill for drying and sweeping. Drying is also aided by heat dissipation from mill draw power which equates to approximately 1 ton moisture per 1000 kWh. Specific power consumption depends upon material hardness and mill efficiency. For ball mills, the range is approximately 10 kWh/ton (mill drive only), for soft, chalky limestone to 25 kWh/ton, for hard materials. For vertical roller mills the range may be 4.5 – 8.5 kWh/ton, and although ID fan power is increased system power is generally about 30% lower than for ball mills. A typical raw mill circuit is shown in the Figure 13.3. As can be seen the preheater exhaust gases (kiln gases) at 290 ^{o}C goes into the mill along with feed material. The material is dried and ground in the mill and fine powder is carried away by kiln gases to multiclones and to bagfilters where the product is collected. The transport of material through the gases is aided by circulating air (CA) fan and dust collector (DC) fan which also help to overcome pressure drop in the system.

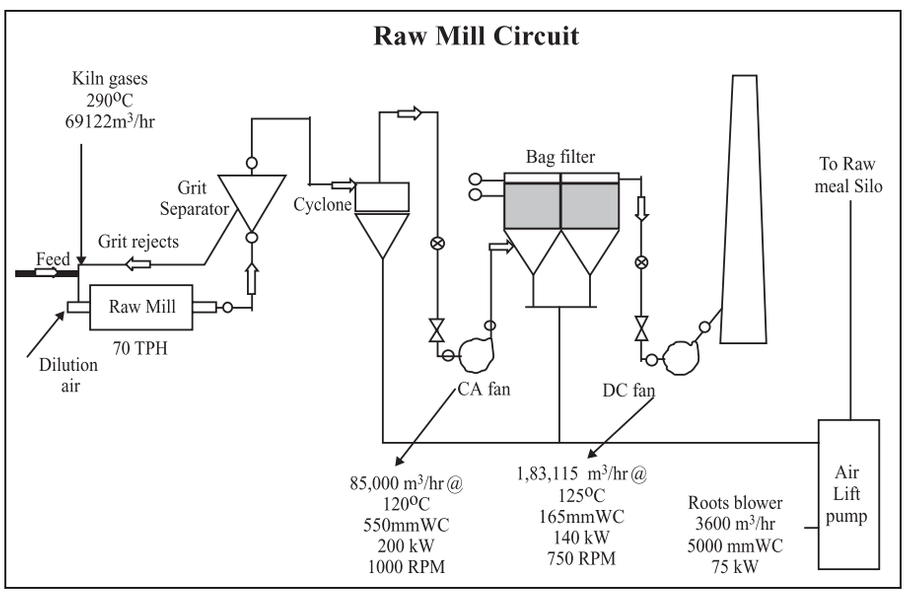

Figure 13.3. Raw Mill Circuit

Raw Mill Heat Balance

It will be useful to perform heat balance of the raw mill to optimize the air flow in the mill to improve productivity besides saving energy. A typical heat balance of raw mill is given in the example below.

Example 13.2.

To determine hot gases required for drying of material in raw mill using heat balance

The hot gases required for the drying of the feed moisture in the raw material while grinding in close circuit ball mill is calculated as below.

Ght	Hot gases temperature	280	^{0}C
Ghs	Specific heat of hot gases	0.34	kcal/ Nm^3 ^{0}C
Rt	Base temperature	20	deg. C
Amb	Ambient air temperature	35	deg. C
Ambs	Specific heat of ambient air	0.30	kcal/Nm^3 ^{0}C
Alt	Altitude (From mean sea level)	950	M
Fq	Fresh feed quantity	100	TPH
Mf	Total fresh feed moisture(surface)	8	%
Mp	Total product moisture (surface)	2	%
Fs	Specific heat of raw material	0.21	kcal/kg ^{0}C
Fa	False air percentage	10	%
P	Power drawn by mill motor	3095	kW
Eg	Amount of dedusting gases	150000	m^3/hr
Egt	Dedusting gases temperature	105	^{0}C
Egs	Specific heat of dedusting gases	0.31	kcal/Nm^3 ^{0}C
Ra	Surface area for Radiation Losses	185	m^2
Rf	Radiation Loss	50	kcal/m^2 ^{0}C difference

Calculations:

Gh Required hot gas quantity (Nm^3/hr)

<u>Heat output:</u>

Step:-1 Heat to raw material

Hop = Fq x Fs x (Egt-Rt-5) kcal/hr

= 100x1000x0.21x (105-20-5)

= 1680000 = 1.68 x 10^6 kcal/hr

Note: The raw material temperature is normally less by 5 deg. C than the exit gas temperature.

Step:-2 Heat to dedusting gases

$$Hog = Eg \times Egs \times (Egt - Rt) \times \frac{273}{Egt + 273} \quad \text{kcal/hr}$$

= 150000 x 0.31x (105-20) x [273/ (105+273)]

= 2854583 = 2.855 x 10^6 kcal/hr

Step:-3 Heat loss due to radiation

Hor = Ra × Rf × (Egt - Rt) kcal/hr

= 185 x 50 x (105-20)

= 786250 = 7.863 x 10^5 kcal/hr

Step:-4 Heat loss to evaporate moisture

How = W × (540 + Egt - Amb) kcal/hr

Moisture to be evaporated,

$$W = \frac{Fq \times (Mf - Mp)}{100 - Mp} \quad \text{kg/hr}$$

= {100 x 1000 x (8-2)}/ (100-2)

= 6122 kg/hr

How = 6122 x {540 + (105 -35)}

= 3734420 = 3.734 x 10^6 kcal/hr

Note: The latent heat of evaporation of water is 540 kcal/kg of water.

Step:-5 Heat loss due to false air

$$Hoa = \frac{Eg \times Fa \times Ambs \times (Egt - Rt)}{100} \quad \text{kcal/hr}$$

= 150000 x 0.1 x 0.30 x (105-20)

= 382500 = 3.825 x10^5 kcal/hr

Total heat loss at output:

Ho = Hop + Hog + Hor + How + Hoa kcal/hr

= 1680000 + 2854583 + 786250 + 3734420 + 382500

Ho = 9437753 = 9.438 x 10^6 kcal/hr

<u>Heat input:</u>

Step:-1 Heat from fresh feed

Hif = Fq × Fs × (Amb - Rt) kcal/hr

= 100 x 1000 x 0.21 x (35-20)

= 315000 = 3.15 x 10^5 kcal/hr

Step:-2 Heat from grinding power

Hip = P × motor efficiency x gear box efficiency x 860

Hip = P × 0.94 x 0.98 x 860 kcal/hr

= 3095 x 0.921 x 860

= 2451000 = 2.451 x10^6 kcal/hr

Step:-3 Heat from false air

$$Hia = \frac{Eg \times Fa \times Ambs \times (Amb - Rt)}{100} \text{ kcal/hr}$$

= 150000 x 0.1 x 0.30 x (35-20)

= 67500 = 6.75 x 10^4 kcal/hr

Total heat input:

Hi = Hif + Hip + Hia kcal/hr

= 315000 + 2451000 + 67500

Hi = 2833500 = 2.83 x 10^6 kcal/hr

Note: In the above calculation, heat input is less than the heat output. Hence it has to be balanced by heat input from hot gases.

To find out the quantity of hot gases required for drying

Heat input (Hi) = Heat output (Ho)

2833500 + (Heat from hot gases) = 9437753

Heat from hot gases,

Gh x Ghs x (Ght – Rt) = (9437753 - 2833500) kcal/hr

Gh x 0.34 x (280-20) = 6604253

Gh = 74709 Nm^3/hr

The result of heat balance is summarized in the Figure 13.4 below.

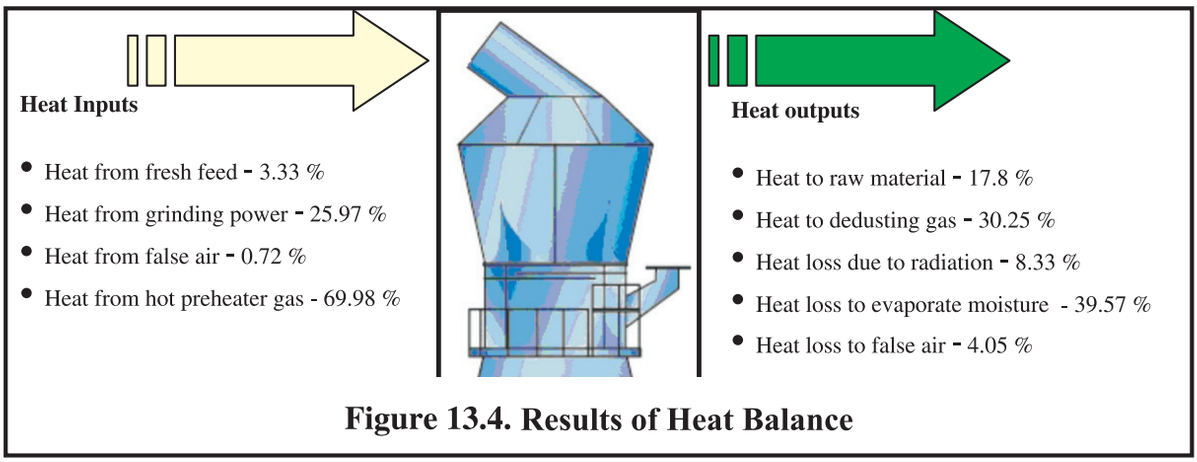

Figure 13.4. Results of Heat Balance

Solved Example:

During heat balance of a 5 stage preheater Kiln in a cement plant, the following data was measured at Preheater (PH) Fan Inlet and clinker cooler vent air fan inlet:

Parameter measured	Temperature	Static Pressure	Avg. Dynamic Pressure	Specific heat	Gas Density at STP	Duct Area
Unit	^oC	(P_s) mm WC	(P_d) mm WC	kcal/kg^oC	kg/m^3	m^2
PH Exit Gas at PH fan Inlet	316	-650	28.6	0.248	1.4	2.27
Clinker cooler vent air at cooler Stack Fan Inlet	268	-56	9.7	0.24	1.29	2.01

Note: Take Pitot tube constant as 0.85, reference temperature 20 ^oC and atmospheric pressure 9908 mm WC.

Other Data:

Clinker Production	Designed specific volume of PH gas	NCV of Coal	Cost of coal	Annual Operation
TPH	Nm^3/kg clinker	kcal/kg	Rs./ton	Hrs.
45.16	1.75	5500	6500	8000

Calculate the following:

i. Specific volume of PH gas as well as cooler vent air (Nm^3/kg clinker)

ii. Heat loss in pre-heater exit gas (kcal/kg clinker)

iii. Heat loss in cooler vent air (kcal/kg clinker)

iv. If the measured specific volume of PH gas (Nm^3/kg clinker) exceeds the design value, calculate the heat loss (kcal/kg clinker) and annual monetary loss due to excessive specific volume of PH gas.

Solution:

i) Density of Pre-heater gas at PH Fan Inlet at prevailing temp., pressure conditions:

$$\rho_{T,P} = \rho_{STP} \text{ X } \frac{273 \text{ X } (9908 + P_S)}{(273 + T) \text{ X } 10334}$$

$$\rho_{T,P} = 1.40 \text{ X } \frac{273 \text{ X } (9908 - 650)}{(273 + 316) \text{ X } 10334} = 0.581 \text{ kg/m}^3$$

Velocity of PH gas

$$v = P_t \sqrt{\frac{2 g P_d}{\rho_{T,P}}}$$

$$v = 0.85 \sqrt{\frac{2 \text{X} 9.8 \text{X} 28.6}{0.581}} = 26.4 \text{ m/sec}$$

Volumetric flow rate of PH gas = velocity X duct cross-sectional area
= 26.4 X 2.27
= 59.9 m^3/sec
= 59.9 X 3600
= 215640 m^3/hr

Specific volume of PH gas = 215640 X 0.58/1.4
= 89491 Nm^3/hr
= 89491/45160 = **1.98 Nm^3/kg clinker**

Similarly density of cooler vent air at cooler vent air fan Inlet at prevailing temperature, pressure conditions:

$$\rho_{T,P} = \rho_{STP} \text{X} \frac{273 \text{X} (9908 + P_S)}{(273 + T) \text{X} 10334}$$

$$\rho_{T,P} = 1.29 \text{X} \frac{273 \text{X} (9908 - 56)}{(273 + 268) \text{X} 10334} = 0.62 \text{ kg/m}^3$$

Velocity of cooler vent air in the fan inlet duct

$$v = P_t \sqrt{\frac{2 g P_d}{\rho_{T,P}}}$$

$$v = 0.85 \sqrt{\frac{2 \text{X} 9.8 \text{X} 9.7}{0.62}} = 14.88 \text{ m/sec}$$

Volumetric flow rate of PH gas = velocity X duct cross-sectional area
= 14.88 X 2.01
= 29.9 m^3/sec
= 29.9 X 3600
= 107640 m^3/hr

Specific volume of cooler vent air = 107640 X 0.62/1.29
= 51734 Nm^3/hr
= 51734/45160 = **1.15 Nm^3/kg clinker**

ii) Heat loss in PH exit gas

Q1 = $m_{ph}\ c_p\ \Delta T$ (C_p of PH gas = 0.248 kcal/kg ^oC)

Q1= 1.98 X 1.4 X 0.248 x (316-20)

= 203.5 kcal/kg clinker

iii) Heat loss in cooler vent air

Q2 = $m_{CA}\ c_p\ \Delta T$ (C_p of cooler vent air = 0.24 kcal/kg ^oC)

Q2 = 1.15 x 1.29 x 0.24 x (268-20)

= 88.3 kcal/kg clinker

iv) Heat Loss due to excess specific volume of PH gas

V_{excess} =1.98 – 1.75 = 0.23 **Nm^3/kg clinker**

Heat loss Q = 0.23 x 1.4 x 0.248 x (316-20) = 23.6 **kcal/kg clinker**

Equivalent coal saving = 23.6/5500 = 0.0043 kg coal/kg clinker or ton of coal/ton of clinker

Coal saving in one hour = 0.0043 x 45.16 = 0.194 TPH

Annual Coal Saving = 0.194 x 8000= 1552 tons of coal per annum

Annual Monetary Saving = 1552 x 6500 = Rs. 100.88 lakhs

<table>
<tr><th colspan="2">QUESTIONS</th></tr>
<tr><td>S-1</td><td>What is the purpose of using preheater exhaust gases in the raw mill?</td></tr>
<tr><td>S-2</td><td>Which compound is the major constituent of clinker?</td></tr>
<tr><td>S-3</td><td>Why is gypsum blended with ground clinker in a cement mill?</td></tr>
<tr><td>S-4</td><td>Name two typical uses of recuperated hot air from clinker cooler</td></tr>
<tr><td>S-5</td><td>Name two areas in the cement plant where majority of thermal energy is used</td></tr>
<tr><td>S-6</td><td>Which is the major Green House Gas evolved during the process of calcinations?</td></tr>
<tr><td>S-7</td><td>Why CO_2 percentage is not used for calculating excess air from cement kilns?</td></tr>
<tr><td>S-8</td><td>What is the maximum temperature encountered in a cement plant and where?</td></tr>
<tr><td>S-9</td><td>List two sources of waste heat from cement plant.</td></tr>
<tr><td>S-10</td><td>What parameters are required for calculating the corrected air density?</td></tr>
<tr><td>L-1</td><td>Chemical analysis of clinker (loss free basis)

<table><tr><td>Constituents Determined</td><td>%</td></tr><tr><td>SiO_2</td><td>22.68</td></tr><tr><td>Fe_2O_3</td><td>5.92</td></tr><tr><td>Al_2O_3</td><td>5.29</td></tr><tr><td>CaO</td><td>63</td></tr><tr><td>MgO</td><td>1.25</td></tr></table>
Calculate heat of formation of clinker?</td></tr>
<tr><td>L-2</td><td>Applying pressure and temperature correction calculate the density of air at a temperature of 120 ^{0}C and a static pressure of – 45 mmWC.</td></tr>
<tr><td>N-1</td><td>From the following data calculate:

i) heat in preheater exhaust gases
ii) heat loss due to incomplete combustion

Data:

Clinker production : 158 TPH
Static pressure : - 682 mmWC
Velocity pressure : 18 mmWC
Diameter of the duct : 3000 mm
Temperature of preheater exhaust gas : 325 ^{0}C
Reference temperature : 20 ^{0}C
Specific heat of preheater exhaust gas : 0.243 kcal/kg ^{0}C

O_2 : 4.0%
CO : 0.3%
CO_2 : 24.1%</td></tr>
</table>

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Chapter 14

Performance Assessment of Buildings and Commercial Establishments

Learning Objectives

In this chapter you will learn about

- The Energy Performance Index (EPI) and Average Annual Hourly EPI.
- The Assessment of cooling load for a building
- The Assessment of HVAC systems
- The Assessment of lighting systems

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Estimate Energy Performance Index (EPI) and Average Annual Hourly EPI (AAhEPI)**
- ✓ **Calculate the cooling load requirement of a building**
- ✓ **Assess the performance of a centralized air conditioning system and its components**
- ✓ **Assess the performance of Lighting System**

14. ENERGY PERFORMANCE ASSESSMENT OF BUILDINGS AND COMMERCIAL ESTABLISHMENTS

14.1 Introduction

The major energy consuming utilities of the buildings and commercial establishments are HVAC, lighting, office systems and other loads. In order to achieve reduction in the energy costs by proper utilization, operation and maintenance of the existing building equipment and utility systems, performance evaluation is required. In addition there is also a need to review the building envelope for minimizing the heat gain and maximizing daylighting usage in the building.

14.2 Purpose of the Performance assessment

The purpose of the performance assessment is to analyse the building for its envelope, load profiles, equipment and system efficiencies and the energy optimization scope for a building by quantifying energy usage. It helps to evaluate the present status, set up a baseline and compare with the standard norms and benchmark.

14.3 Determining Energy Performance Index (EPI) and Average Annual Hourly EPI (AAHEPI)

The different sources of energy used in the building have to be identified with the help of facility people and walkthrough survey of the facilities.

Step 1 -Data Collection

The total electrical energy consumption of the facility both Grid and Captive is to be collected month wise over a one year period.

The total built up area of the building needs to be collected. The built up area is the carpet area plus area covered by thickness of walls plus balconies etc., and would exclude parking areas.

The operating hours per day and working days per year have to be collected.

Step 2 -Analysis

Energy performance Index (EPI): Energy Performance Index is a measuring tool to evaluate the performance of the building in terms of the total energy consumption and the total built up area.

$$Energy\ Performance\ Index\,(EPI), kWh/annum/m^2 = \frac{(EB\ energy + DG\ energy),\ kWh/annum}{Total\ builtup\ area, m^2}$$

Average Annual Hourly Energy Performance Index (AAHEPI): The number of operating hours of a building may vary depending on the nature of operation. The AAHEPI enables the performance assessment taking into account the actual operating hours which will be more meaningful. AAHEPI is defined as follows

$$AAHEPI, Wh/h/m^2 = \frac{EPI\ ,\ kWh/annum/m^2}{Operating\ hours\ per\ annum} \times 1000$$

Example 14.1: Estimation of EPI and AAHEPI

A commercial building located in warm and humid climatic region, has a built up area of 17814 m^2. The annual energy consumption of the building is 3735640 kWh from the Grid and 2964758 kWh from captive DG sets. The facility operates 12 hrs a day and 250 days in a year. Calculate the EPI and AAHEPI.

Solution:

$$\text{Energy Performance Index (EPI), kWh/annum/m}^2 = \frac{(3735640+2964758),\ \text{kWh/annum}}{17814, \text{m}^2}$$

$$\text{Energy Performance Index (EPI), kWh/annum/m}^2 = 376$$

$$\text{AAHEPI, Wh/h/m}^2 = \frac{376,\ \text{kWh/annum/m}^2}{12 \times 250} \times 1000$$

$$\text{AAHEPI, Wh/h/m}^2 = 125$$

14.4 Significance of Building Envelope

The building envelope is the interface between the interior of the building and outdoor environment including the walls, roof and foundation. By acting as a thermal barrier, the building envelope plays an important role in regulating interior temperatures and helps in determining the amount of energy required to maintain thermal comfort. The envelope design and orientation also determines the amount of artificial lighting required in the building.

Envelope design basics

The energy performance of a building envelope is influenced by number of factors. For example, these may include design elements such as the physical orientation of the building and the amount of sunlight that penetrates into the interior work spaces. Other factors may also include heat transfer characteristics (both losses and gains) and location of the building envelope components such as walls, windows, doors, floors and roofs. The energy performance of the building may also be influenced by any natural air infiltration through the building envelope (e.g. cracks). The sources of heat gain into the building space are depicted in figure 14.1.

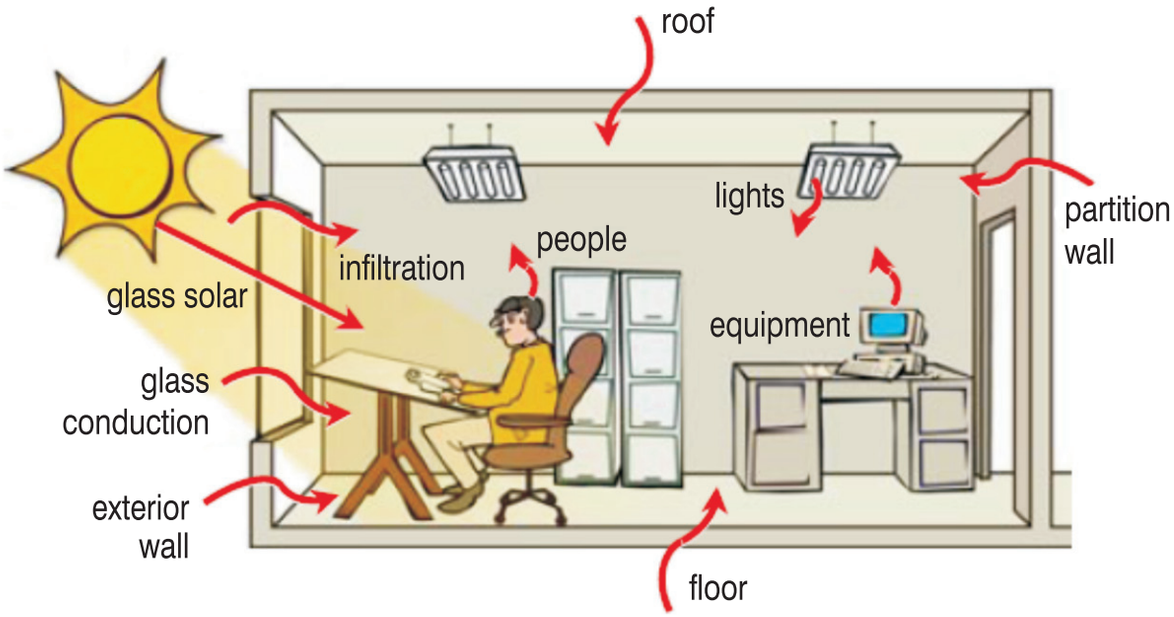


Figure 14.1 Sources of heat gain into the building space

An integrated building design considers the envelope, the heating, ventilation, cooling (HVAC) system and the lighting system as a whole, rather than dealing with these independently. Changing the specifications of one system can affect the performance of the other two significantly. For instance, investments in good insulation of the roof, energy-efficient windows or increased envelope air tightness can result in a smaller HVAC system, thereby reducing its first cost as well as recurring energy cost. Similarly, an inefficient lighting system not only increases lighting energy consumption but could also increases the cooling load on HVAC system, thereby increasing the energy consumption further.

When a building is in cooling mode, solar heat gains need to be minimized within the building space while optimizing daylight and intake of outside air. Outside air could be introduced particularly during evening/night hours when the ambient temperature drops. This strategy cools the thermal mass in the building during night hours and reduces overall cooling load during the next day. On the other hand, if the building is in a heating mode, the envelope needs to be designed with appropriate glazing selection coupled with shading strategy to enhance solar heat gains during daytime. Therefore in practice, the architects and building designers need to integrate and balance these varying requirements considerations while designing an energy-efficient building.

14.5 Assessment of HVAC Systems

The detailed layout of HVAC plant should be collected along with the design parameters of all the HVAC equipment. The main objective of the assessment will be to determine the actual operating refrigeration load (TR), measure the actual power drawn and compare the actual kW/TR with the design.

The operating parameters of Chiller plant has to be monitored for one day. The temperature measurements with 0.1 least count have to be carried out periodically for chilled water and cooling water. The flow

measurements are to be carried out with an ultrasonic flow meter. Simultaneous power measurements need to be done with the compressor motor, chilled water, cooling water pumps and cooling tower fans/condenser fans. The detailed calculation will have to be performed for various equipment/systems as given in the table 14.1 below.

Table 14.1 Performance Calculations Reference

Sl.No.	Equipment/system	Reference
1	Chiller	Chapter 4.9
2	Pump	Chapter 4.7
3	Fans	Chapter 4.6
4	Motors	Chapter 4.5
5	AHU	Chapter 4.9
6	Cooling tower	Chapter 3.7

If further in-depth assessments need to be performed for calculating the prevailing air conditioning load, the following method may be pursued. This method merely highlights the concepts and intensity behind the cooling load calculations. The detailed calculations are outside the scope of this book as they are heavily dependant on vast data from ASHRAE hand book.

Terminology used in estimating cooling load

Commonly used terms relative to heat transmission and load calculations are defined below in accordance with ASHRAE Standard 12-75, Refrigeration Terms and Definitions.

Space – is either a volume or a site without a partition or a partitioned room or group of rooms.

Room – is an enclosed or partitioned space that is usually treated as single load.

Zone – is a space or group of spaces within a building with heating and/or cooling requirements sufficiently similar so that comfort conditions can be maintained throughout by a single controlling device.

Cooling Load Temperature Difference (CLTD) – an equivalent temperature difference used for calculating the instantaneous external cooling load across a wall or roof.

Sensible Heat Gain – is the energy added to the space by conduction, convection and/or radiation.

Latent Heat Gain – is the energy added to the space when moisture is added to the space by means of vapor emitted by the occupants, generated by a process or through air infiltration from outside or adjacent areas.

Radiant Heat Gain – the rate at which heat absorbed is by the surfaces enclosing the space and the objects within the space.

Space Heat Gain – is the rate at which heat enters into and/or is generated within the conditioned space during a given time interval.

Space Cooling Load – is the rate at which energy must be removed from a space to maintain a constant space air temperature.

Thermal Transmittance or Heat Transfer Coefficient (U-factor) – is the rate of heat flow through a unit area of building envelope material or assembly, including its boundary films, per unit of temperature difference between the inside and outside air. The U-factor is expressed in W/ m^2 ^oK)

Thermal Resistance (R) – is the reciprocal of a heat transfer coefficient and is expressed in m^2 ^oK/W. The value of R is also used to represent Thermal resistivity, the reciprocal of the thermal conductivity.

Types of loads

There are two distinct components of the air conditioning load;

1. The sensible load (heat gain)
2. The latent load (water vapor gain).

Sensible Loads: Sensible heat gain is the direct addition of heat to a space, which shall result in increase in space temperatures. The factors influencing sensible cooling load:

1. Solar heat gain through building envelope (exterior walls, glazing, skylights, roof, floors, over crawl space)
2. Partitions (that separate spaces of different temperatures)
3. Ventilation air and air infiltration through cracks in the building, doors, and windows
4. People in the building
5. Equipment and appliances operated in the summer
6. Lights

Latent Loads: Latent heat gain is the heat contained in water vapor. Latent heat does not cause a temperature rise, but it constitutes a load on the cooling equipment Latent load is the heat that must be removed to condense the moisture out of the air.

The sources of latent heat gain are:

1. People (breathing)
2. Cooking equipment
3. Housekeeping, floor washing etc.

4. Appliances or machinery that evaporates water
5. Ventilation air and air infiltration through cracks in the building, doors, and windows

The total cooling load is the summation of sensible and latent loads.

Cooling Loads Classified by Inside-Outside Environment

Buildings can be classified as envelope-load-dominated and interior-load-dominated. The envelope heat flows are termed as "external loads", in that they originate with the external environment. The other loads are termed as "Internal loads", in that they are generated from within the building itself. The percentage of external versus internal load varies with building type, site climate, and building design decisions. It is useful to identify whether internal or external loads will dominate a building, as this information should substantially change the focus of design efforts related to control and energy efficiency.

External Loads

External cooling loads consist of the following:

1. Sensible loads through opaque envelope assemblies (roofs, walls, floors)
2. Sensible loads through transparent or translucent envelope assemblies (skylights, windows, glazed openings)
3. Sensible loads through ventilation and infiltration (air leakage)
4. Latent loads through ventilation and infiltration.

Internal Loads

1. Typical sources of internal heat gain are people, lights, cooking processes, and other heat-generating equipment, such as motors, appliances, and office equipment.
2. While all of these sources contribute sensible heat to the space, people, cooking processes, and some appliances (such as a coffee maker) also contribute latent heat to the space.

CLTD/SCL/CLF Method of Load Calculation (ASHRAE Handbook Fundamentals 1997)

The heat gain in to the building is not converted to cooling load instantaneously.

CLTD (cooling load temperature difference),
SCL (solar cooling load factor), and
CLF (cooling load factor): all include the effect of

(1) time-lag in conductive heat gain through opaque exterior surfaces and
(2) time delay by thermal storage in converting radiant heat gain to cooling load.

This approach allows cooling load to be calculated manually by use of simple multiplication factors.

a. **CLTD** is a theoretical temperature difference that accounts for the combined effects of inside and outside air temp difference, daily temp range, solar radiation and heat storage in the construction assembly/building mass. It is affected by orientation, tilt, month, day, hour, latitude, etc. *CLTD factors are used for adjustment to conductive heat gains from walls, roof, floor and glass.*

b. **CLF** accounts for the fact that all the radiant energy that enters the conditioned space at a particular time does not become a part of the cooling load instantly. The CLF values for various surfaces have been calculated as functions of solar time and orientation and are available in the form of tables in ASHRAE Handbooks. *CLF factors are used for adjustment to heat gains from internal loads such as lights, occupancy, power appliances.*

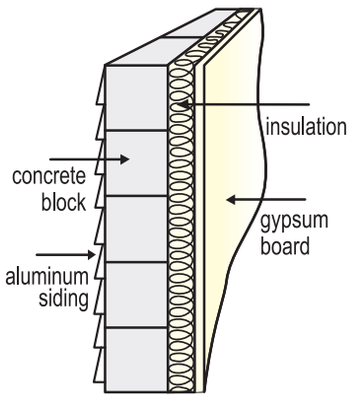
c. **SCL** factors are used for adjustment to transmission heat gains from glass.

Example – 14.2 Assessing cooling load using CLTD/SCL/CLF Method (ASHRAE Handbook Fundamentals 1997)

The method is explained alongside with an example.

Determine the cooling load from the following data:

Total area of wall + windows	- 50.7 m^2
Desired indoor conditions:	
Dry-bulb temperature	- 25.6°C
Relative humidity	- 50%
Details of the wall:	
Resistance of aluminum siding	- 0.11 m^2 °K/W
Resistance of lightweight concrete block (200 mm)	- 0.35 m^2 °K/W
Resistance of fiberglass insulation (90 mm)	- 2.29 m^2 °K/W
Resistance of gypsum board (12.7 mm)	- 0.08 m^2 °K/W



Additionally, there is a film of air on the outside surface of the wall (R - 0.044 m^2 °K/W), assuming air moving at 12 km/hr during the summer and another film of air on the inside surface of the wall (R - 0.12 m^2 °K/W), assuming still air.

Eight clear double-pane windows mounted in aluminum frames. Each window is 1.2 m x 1.5 m.

Flat 13.7 m x 18.3 m roof constructed of 100 mm concrete with 90 mm insulation and steel decking.

Space is occupied from 8:00 to 1700 hrs by 18 people doing moderately active work.

Fluorescent lighting in space = 21.5 W/m^2.

Computers and office equipment in space = 5.4 W/m^2 plus one coffee maker.

U-Factor of wall:

The U-factor of this wall is calculated by adding the thermal resistances of each of these layers and then taking the inverse.

$$U = \frac{1}{R_{\underset{\text{air film}}{\text{outdoor}}} + R_{\text{siding}} + R_{\underset{\text{block}}{\text{concrete}}} + R_{\text{insulation}} + R_{\underset{\text{board}}{\text{gypsum}}} + R_{\underset{\text{air film}}{\text{indoor}}}}$$

$$U = \frac{1}{0.04 + 0.11 + 0.35 + 2.29 + 0.08 + 0.12} = \frac{1}{2.99} = 0.33 \text{ W/m}^2 \bullet {}^\circ\text{K}$$

U-Factor of roof:

The U-factor of the roof in this example is calculated in a similar manner.

$$U = \frac{1}{R_{\underset{\text{air film}}{\text{outdoor}}} + R_{\underset{\text{roofing}}{\text{built up}}} + R_{\text{insulation}} + R_{\underset{\text{concrete}}{\text{lightweight}}} + R_{\underset{\text{decking}}{\text{metal}}} + R_{\underset{\text{air film}}{\text{indoor}}}}$$

$$U = \frac{1}{0.04 + 0.06 + 2.29 + 0.54 + 0 + 0.16} = \frac{1}{3.09} = 0.323 \text{ W/m}^2 \bullet {}^\circ\text{K}$$

I. External heat gain

i) Conduction heat gain through the wall (if assumed to be shaded at all times)*:

U-factor	- 0.33 W/m^2 °K
Total area of wall + windows	- 50.7 m^2
Area of windows	- 14.4 m^2
Net area of wall	- 36.3 m^2
Outdoor temperature	- 35 °C
Indoor temperature	- 25.6 °C

$$Q = U \times A \times \Delta T$$

$$Q = 0.33 \times 36.3 \times (35 - 25.6) = 113 \text{ W}$$

** The assumption that the surface is completely shaded does not account for the additional heat gain that occurs when the sun shines on a surface. Solar heat, therefore, must be considered, as it*

constitutes an important part of the total cooling load of most buildings.
A factor called the cooling load temperature difference (CLTD) is used to account for the added heat transfer due to the sun shining on exterior walls, roofs, and windows and the capacity of the wall and roof to store heat. The CLTD is substituted for ΔT in the equation to estimate heat transfer by conduction.

$$Q = U \times A \times CLTD$$

ASHRAE tables contain CLTD factors for walls and roofs. The data relevant to a place/location can be taken from the table. It should be noted that the data in this table are based on the following assumptions:

Indoor air	- 25.6 °C
Maximum outdoor air	- 35 °C
Average outdoor daily temperature range	- 11.7 °C
Month	- 21st July
Latitude	- 40° North
Surface	- Dark-colored

Tables for various wall and roof types, as well as correction factors for applications that differ from these assumptions can be found in the ASHRAE Handbook - Fundamentals and ASHRAE's Cooling and Heating Load Calculation Principles manual.

The wall used in this example is classified as wall type 9. At 1700 hours, the CLTD for a wall of this type is 12 °C. This means that, even though the actual dry-bulb temperature difference is only 9.4 °C (35°C - 25.6°C), the sun shining on the outer surface of this wall increases the "effective temperature difference" to 12 °C.

It can be found from the ASHRAE tables that the CLTD increases later in the day, and then begins to decrease in the evening as the stored heat is finally transferred from the wall into the space.

Using CLTD instead of ΔT, the heat gain by conduction, through the wall and the roof is to be determined. The U-factors are the same as calculated above.

Conduction heat gain through the wall:

U-factor	- 0.33 W/m^2 °K
Net area of wall	- 36.3 m^2
CLTD at 17 hours	- 12 °C

$$Q = 0.33 \times 36.3 \times 12 = 144 \text{ W}$$

ASHRAE tables include CLTD factors for several types of roofs. The roof in this example building is classified as Roof Type 2. The data in ASHRAE tables are based on assumptions similar to the CLTD table for walls. At Hour 17, the CLTD for a flat roof of this type is 44 °C.

ii) Conduction heat gain through the roof:

U-factor - 0.323 W/m^2 °K
Area of roof - 13.7 m x 18.3 m = 250.7 m^2
CLTD at 17 hours - 44 °C

Q = 0.323 x 250.7 x 44 = 3,563 W

iii) Conduction heat gain through the windows:

Estimating the heat gain by conduction through a window is very similar to walls and roofs.

The ASHRAE Handbook- 1997 Fundamentals includes U-factors for common window assemblies.

The windows in this example are double-pane windows with a 6.4 mm air space between the panes. Assuming that the windows are fixed (not operable) with aluminum frames and a thermal break, the U-factor is 3.56 W/m^2 °K.

Using this U-factor, the heat gain by conduction, through the eight windows is to be determined. The ASHRAE table 34 includes CLTD factors for glass. It is also based on similar assumptions as the CLTD tables for walls and roofs. At Hour 17, the CLTD for a glass window is 7 °C.

$Q_{windows}$ - 3.56 x 14.4 x 7 = 359 W

While glass windows of double- or triple-pane construction do an excellent job of reducing heat transfer by conduction, they do not appreciably reduce the amount of solar radiation directly into a space. To limit the amount of solar radiation entering the space, heat-absorbing glass, reflective glass, or internal or external shading devices can be used.

iv) Solar radiation through glass

The equation used to predict the solar heat gain through glass is

Q = A x SC x SCL

Where,

Q - Heat gain by solar radiation through glass, W
A - Total surface area of the glass, m^2
SC - Shading coefficient of the window, dimensionless
SCL - Solar cooling load factor, W/m^2

The solar cooling load (SCL) factor is used to estimate the rate at which solar heat energy radiates directly into the space, heats up the surfaces and furnishings and is later released to the space as a sensible heat gain. Similar to CLTD, the SCL factor is used to account for the capacity of the space to absorb and store heat.

The value of SCL is based on several variables, including the direction that the window is facing, time of day, month, and latitude. These four variables define the angle at which the sun's rays strike the surface of the window. The next two variables, the construction of the interior partition walls and the type of floor covering, help define the capacity of the space to store heat. This affects the time lag between the time that the solar radiation warms up the space furnishings and the time that the heat is released into the space. The last variable, whether or not internal shading devices are installed, affects the amount of solar heat energy passing through the glass.

The 1997 ASHRAE Handbook-Fundamentals contains tables of SCL values for common space types, based on the combinations of these variables. ASHRAE table 36 contains SCL factors for a space type similar to the one in this example. The space in this example is classified as Space Type A. The data in this table is based on 21st July and 40° North latitude. At 17 hours, the SCL for the windows in this example space is 605 W/m^2.

The shading coefficient (SC) is an expression used to define how much of the radiant solar energy, that strikes the outer surface of the window, is actually transmitted through the window and into the space. The shading coefficient for a particular window is determined by comparing its reflective properties to a standard reference window. The ASHRAE table includes shading coefficients for common window systems. When the value for the shading coefficient decreases, more of the sun's rays are reflected by the outer surface of the glass.

The windows in this example space are constructed of two panes of 6.4 mm clear glass with an air space between the panes. The glass is mounted in an aluminum frame and the windows are fixed (not operable). The SC for this type of window from ASHRAE table is 0.74. The heat gain by solar radiation through the windows on the wall of this example space can be determined as follows:

Solar radiation heat gain through the windows on the wall:

Total area of glass	- 8 x (1.2 m x 1.5 m) = 14.4 m^2
SC	- 0.74
SCL at 17 hours	- 605 W/m^2

Q = 14.4 x 0.74 x 605 = 6,447 W

II. Internal Heat Gains

v) Heat gain from people

People generate more heat than is needed to maintain body temperature. This surplus heat is dissipated to the surrounding air in the form of sensible and latent heat. The amount of heat released by the body varies with age, physical size, gender, type of clothing, and level of physical activity.

The equations used to predict the sensible and latent heat gains from people in the space are:

Q_s = Number of people x Sensible heat gain/person x CLF
Q_L = Number of people x Latent heat gain/person

Where,

Q_s - Sensible heat gain from people, W
Q_L - Latent heat gain from people, W
CLF - Cooling load factor, dimensionless

Similar to the use of the CLTD for conduction heat gain and SCL for solar heat gain, the cooling load factor (CLF) is used to account for the capacity of the space to absorb and store heat. Some of the sensible heat generated by people is absorbed and stored by the walls, floor, ceiling, and furnishings of the space and released at a later time. Similar to heat transfer by conduction through an external wall, the space can therefore experience a time lag between the time that the sensible heat is originally generated and the time that it actually contributes to the space cooling load. For heat gain from people, the value of CLF depends on

- The construction of the interior partition walls in the space
- The type of floor covering
- The total number of hours that the space is occupied and
- The number of hours since the people entered the space.

Tables from ASHRAE Handbook of fundamentals give the CLF values. For example if CLF is 0.65 it means that 35% (1- 0.65) of the sensible heat gain from the people is absorbed by the surfaces and furnishings in the space and 65% is the actual cooling load in the space. As the people are in the space for a longer period of time, the surfaces and furnishings of the space can no longer absorb as much heat and they release the heat that was absorbed earlier in the day. For example, if the people enter the space at 8 a.m and remain for a period of 8 hours, at 2 p.m. (6 hours after entering), the CLF is 0.91 i.e 91% of the sensible heat gain from the people is seen as a cooling load in the space. Only 9% is absorbed by the surfaces and furnishings of the space.

If the space is not maintained at a constant temperature during the 24-hour period, however the CLF is assumed to equal 1.0. Most air-conditioning systems designed for non-residential buildings either shut the system off at night or raise the temperature set point to reduce energy use. Thus, it is uncommon to use a CLF other than 1.0 for the cooling load due to people.

The internal heat gain from people in this example space can be determined based on the ASHRAE tables. From the tables it is observed that people participating in moderately active office work generate 75 W sensible heat and 55 W latent heat.

Internal heat gain from people:

Number of people - 18
Sensible heat gain/person - 75 W
Latent heat gain/person - 55 W
CLF - 1.0
(because the space temperature set point is increased at night)

Q_s = 18 people x 75 W per person x 1.0 = 1,350 W

Q_L = 18 people x 55 W per person = 990 W

vi) Heat gain from lighting

Heat generated by lights in the space is a significant contribution to the cooling load. For example, a 120-watt light fixture generates 120 W of heat—approximately the same amount of heat gain generated by an average office worker. Additionally, when estimating the heat gain from fluorescent lights, approximately 20% is added to the lighting heat gain to account for the additional heat generated by the ballast.

The equation used to estimate the heat gain from lighting is:

Q = watts x ballast factor x CLF

Where,

Q - Sensible heat gain from lighting, W
Watts - Total energy input to lights, W
Ballast factor - 1.2 for fluorescent lights, 1.0 for incandescent lights
CLF - Cooling load factor, dimensionless

Similar to the sensible heat gain from people, a cooling load factor (CLF) can be used to account for the capacity of the space to absorb and store the heat generated by the lights. If the lights are left on 24 hours a day, or if the air-conditioning system is shut off or set back at night, the CLF is assumed to be equal to 1.0.

The internal heat gain from lighting in this example space can be determined as follows.

Amount of lighting in space	- 21.5 W/m^2	
Floor area	- 13.7 m x 18.3 m	= 250.7 m^2
Total lighting energy	- 21.5 W/m^2 x 250.7 m^2	= 5,400 W
Ballast factor	- 1.2 (fluorescent lights)	
CLF	- 1.0	

(because the space temperature set point is increased at night)

$$Q = 5,400 \times 1.2 \times 1.0 = 6,480 \text{ W}$$

vii) Heat generated by equipment

There are many types of appliances and equipment in restaurants, schools, office buildings, hospitals, and other types of buildings. This equipment may generate a significant amount of heat and should be accounted for when estimating the space cooling load.

The ASHRAE handbook contains tables of sensible and latent heat gains from various types of office and restaurant equipment, although data for the actual piece of equipment is preferred, if available.

Using this table, it is found that the coffee maker contributes 1,050 W of sensible heat and 450 W of latent heat to this example space. Additionally, there are 5.4 W/m^2 of computers and other office equipment in the space.

Therefore, the internal heat gain from computers and office equipment is:

Sensible heat gain = 5.4 W/m^2 x 250.7 m^2 = 1,354 W

Similar to the sensible heat gain from people and lighting, tables of cooling load factors (CLF) can be used to refine this estimate. If the equipment is left on 24 hours a day, or if the air-conditioning system is shut off or set back at night, the CLF is assumed to be equal to 1.0. In this example, the CLF is 1.0 because the space temperature set point is increased at night.

viii) Heat gain through infiltration

In a typical building, air leaks into or out of a space through doors, windows, and small cracks in the building envelope. Air leaking into a space is called infiltration. During the cooling season, when air leaks into a conditioned space from outdoors, it can contribute to both sensible and latent heat gain in the space, because the outdoor air is typically warmer and more humid than the indoor air.

Before estimating the heat gain from infiltration, we must first estimate the amount of air that is leaking into the space. This is done through the air change method.

It involves estimating the number of air changes per hour that can be expected in spaces of a certain construction quality. Using this method, the quantity of infiltration air is estimated using the equation:

Infiltration airflow = (volume of space x air change rate) / 3,600

Where,

Infiltration airflow - Quantity of air infiltrating into the space, m^3/s
Volume of space - Length x width x height of space, m^3
Air change rate - Air changes per hour
3,600 - Conversion from hours to seconds

ASHRAE tables provide estimates for infiltration using the air change method. Assuming that the space in this example is of average construction and kept at a positive pressure relative to the outdoors, air changes/hr of infiltration is estimated to be 0.3.

Volume of space = 13.7 m x 18.3 m x 3.7 m = $927.6\ m^3$
Infiltration airflow = 927.6 x 0.3/ 3600 = $0.077\ m^3/s$

The equation used to estimate the sensible heat gain from infiltration is:

$$Q_S = 1{,}210 \times \text{airflow} \times \Delta T$$

Where,

Q_s - Sensible heat gain from infiltration, W
1,210 - Product of density and specific heat, $J/m^3\ °K$
Airflow - Quantity of air infiltrating the space, m^3/s
ΔT - Design outdoor dry-bulb temperature minus the desired indoor dry-bulb temperature, °C

The equation used to estimate the latent heat gain from infiltration is:

$$Q_L = 3{,}010 \times \text{airflow} \times \Delta h$$

Where,

Q_L - Latent heat gain from infiltration, W
3,010 - Latent heat factor, J kg dry air /m^3 grams of water
Airflow - Quantity of air infiltrating the space, m^3/s
Δh - Design outdoor humidity ratio minus the desired indoor humidity ratio, grams of water/kg of dry air

The psychrometric chart can be used to determine the humidity ratio for both outdoor and indoor conditions.

Heat gain from infiltration can be calculated as follows:

Infiltration airflow - 0.077 m^3/s

Outdoor conditions
Dry bulb temp. - 35 °C
Wet bulb temp. - 25 °C
h_o - 15 grams of water/kg dry air
Indoor conditions
Dry bulb temp. - 25.6 °C
Relative humidity - 50%
h_i - 10 grams of water/kg dry air

Q_s = 1,210 x 0.077 x (35 - 25.6) = 876 W
Q_L = 3,010 x 0.077 x (15 - 10) = 1,159 W

The figures 1,210 and 3,010 in the above equations are not constants, but are derived from properties of air at "standard" conditions (21°C dry air at sea level). Air at other conditions and elevations will cause these factors to change.

Summary of space cooling loads

S.No.	Space load components	Sensible heat load, W	Latent head load, W
i	Conduction through exterior wall	144	
ii	Conduction through roof	3,563	
iii	Conduction through windows	359	
iv	Solar radiation through windows	6,447	
v	People	1,350	990
vi	Lights	6,480	
vii	Equipment	2,404	450
viii	Infiltration	876	1,159
	Total space cooling load	**21,623**	**2,599**

This completes the estimation of the components of the cooling load for the space. The total space cooling load will be used during a simplified psychrometric analysis in Period Three to determine the quantity and temperature of air required to condition this space.

III. Cooling load due to ventilation

In addition to these space cooling loads, there are other loads that affect the cooling coil in the building HVAC system. These include the load of the outdoor air, deliberately brought into the building for ventilation purposes and heat generated by the fans in the system. These loads are added to the space load to determine the total cooling load for the building. Estimating these additional components is necessary to properly size the cooling coil for the system.

The amount of outdoor air required for a space is often prescribed by local building codes or industry standards. One such standard, ASHRAE Standard 62, Ventilation for Acceptable Air Quality, prescribes the quantity of outdoor air required per person (or per unit area) to provide adequate ventilation for various types of spaces.

In this example, calculating the required quantity of outdoor air involves multiplying the number of people in the space by the 0.01 m^3/s of outdoor air required per person in an office space.

Ventilation airflow = 18 people x 0.01 m^3/s / person = 0.18 m^3/s

The sensible and latent loads from ventilation are calculated using the same equations as for infiltration:

$$Q_s = 1{,}210 \times \text{airflow} \times \Delta T$$

$$Q_L = 3{,}010 \times \text{airflow} \times \Delta h$$

Cooling load due to the conditioning of ventilation air:

Ventilation airflow - 0.18 m^3/s
Outdoor conditions:
T_o - 35 °C
h_o - 15 grams of water/kg dry air
Indoor conditions:
T_i - 25.6 °C
h_i - 10 grams of water/kg dry air

Q_s = 1,210 x 0.18 x (35 - 25.6) = 2,047 W

Q_L = 3,010 x 0.18 x (15 - 10) = 2,709 W

IV. System heat gains

There may be other sources of heat gain within the HVAC system. One example is heat generated by supply air fans. The heat gain from fans is associated with three energy conversion losses as given below.

Fan motor heat gain = Power input to motor x (1-Motor efficiency)

Fan blade heat gain = Power input to fan x (1- Fan efficiency)

Duct heat gain = Power input to fan x Fan efficiency

The system heat gains are not considered in this example calculation

Summary of total coil cooling loads

Coil load components	Sensible load, W	Latent load, W
Conduction through roof	3,563	
Conduction through exterior wall	144	
Conduction through windows	359	
Solar radiation through windows	6,447	
People	1,350	990
Lights	6,480	
Equipment	2,404	450
Infiltration	876	1,159
Total space cooling load	21,623	2,599
Ventilation	2,047	2,709
Total coil cooling load	**23,670**	**5,308**

Calculation of supply air

The proportions of sensible and latent heat must be known in order to determine the proper condition of the air supplied to cool the space. This Sensible Heat Ratio (SHR), is the ratio of sensible heat gain to total (sensible + latent) heat gain, and is defined as follows:

SHR = Sensible heat gain / (Sensible heat gain + Latent heat gain)

= 21623 W / (21623 W + 2599 W)

= 0.89

This means in this example 89% of the cooling load for the space under consideration is sensible and 11% is latent.

Supply air flow

= Sensible heat gain / {1210 x (Room dry bulb temp. – Supply dry bulb temp.)}

The next step is to either assume the supply air dry bulb temperature and calculate the supply air flow or assume the supply air flow and calculate the supply air temperature.

In this example assuming supply air dry bulb temperature as 12.8 ^{0}C

Supply air flow = 21623 W/{1210 x (25.6 ^{0}C – 12.8 ^{0}C)}

= 1.4 m^3/s

14.6 Performance Assessment of Lighting System

Lighting accounts for a significant portion of the energy use in commercial buildings. In addition, heat generated by lighting contributes to the thermal load to be removed by the cooling equipment. The details of the lighting source which include the wattage, type of lamp and inventory of lamp needs to be collected floor wise or section wise and lighting power density (LPD) is to be determined.

Lighting Power Density (LPD): It is defined as the ratio of total operating lighting load (W) for a specified region of the building to the built up area (m^2) of that specified region. LPD for a typical building is assessed and given in the table 14.2.

Table 14.2 Determination of lighting power density (LPD)

Floor reference	Number of lamps	Wattage of Lamp	Types of lamp	Total Lighting Load in kW	Total Area in m^2	Lighting power density in W / m^2
Floor- 1	165	36	CFL	17.80	2120	8.40
Floor - 2	212	36	CFL	22.86	2722	8.40
Floor – 3	568	36	CFL	61.32	4247	14.44
Admin	83	36	CFL	5.98	2132	2.80
Amenities & Cafeteria	260	36	CFL	9.36	3549	2.64

Methodology of Lighting System Assessment:

Calculation of Installed Load Efficacy Ratio (ILER):

$$\text{Installed Load Efficacy Ratio (ILER)} = \frac{\text{Actual, lux/W/m}^2}{\text{Target, lux/W/m}^2}$$

$$\text{Room Index (RI)} = \frac{L \times W}{H_m \times (L + W)}$$

Where, L = length of interior; W = width of interior; Hm = the mounting height, which is the height of the lighting fittings above the horizontal working plane.

Determination of Measurement Points

The number points in which the lux level measurements are to be made depends on room index. The table 14.3 gives the minimum number of measurement points for a given room index.

Table 14.3 Number of lux measurement points

Room Index	Minimum number of measurement points
Below 1	9
1 and below 2	16
2 and below 3	25
3 and above	36

The average maintained illuminance is obtained by averaging the lux readings taken at different measurement points in a room or floor. The target lux/W/m^2 for maintained illuminance on horizontal plane is given in table 14.4.

Table 14.4 Target lux/W/m² values for maintained illuminance on horizontal plane for all room indices and applications

Room Index	Commercial lighting (Offices, Retail stores, etc.) & very clean industrial applications, Standard or good colour rendering. Ra: 40-85	Industrial lighting (Manufacturing areas, Workshops, Warehousing etc.) Standard or good colour rendering. Ra: 40-85	Industrial lighting installations where standard or good colour rendering is not essential but some colour discrimination is required. Ra: 20-40
5	53	49	67
4	52	48	66
3	50	46	65
2.5	48	44	64
2	46	42	61
1.5	43	39	58
1.25	40	36	55
1	36	33	52

Ra : Colour Rendering Index

The installed load efficacy ratio is the ratio of actual calculated lux/W/m² to the target lux/W/m² given in the table 14.5.

Table 14.5: ILER Calculation Procedure:

STEP 1	Measure the floor area of the interior	Area = 45 m²
STEP 2	Calculate the Room Index	RI = 1.93

STEP 3	Determine the total circuit watts of the installation by a power meter if a separate feeder for lighting is available. If the actual value is not known a reasonable approximation can be obtained by totaling up the lamp wattages including the ballasts	Total circuit watts = 990W
STEP 4	Calculate Watts per square metre, **3 ÷1**	W/m² = 22
STEP 5	Ascertain the average maintained illuminance, Eav. Maintained (average lux levels measured at 18 points)	E_{av}.maint. = 700
STEP 6	Divide **5** by **4** to calculate the actual lux per watt per square Metre	Lux/W/m² = 31.8
STEP 7	Obtain target Lux/W/m² lux for type of the type of interior/application and RI **(**2): (Refer Table 14.4)	Target Lux/W/m² = 46
STEP 8	Calculate **I**nstalled **L**oad **E**fficacy **R**atio **(6 ÷ 7)**	ILER = 0.7

Referring to Table 14.6, ILER of 0.7 means that there is a scope for review of the lighting system.

Table 14.6 Assessment (Indicators of Performance)

ILER	Assessment
0.75 or over	Satisfactory to Good
0.51 – 0.74	Review suggested
0.5 or less	Urgent action required

Having calculated the ILER for an existing lighting installation, then the difference between the actual ILER and the best possible (1.0) can be used to estimate the energy wastage.

Annual Energy Wastage : (1 – ILER) x (Watts / 1000) x operating hours per annum

: (1 – 0.7) x (990 / 1000) x 8 hrs/day x 300 days

: 712 .8 kWh / annum

Inference: ILER Ratios of 0.75 or more may be considered to be satisfactory. Existing installations with ratios of 0.51 - 0.74 certainly merit investigation to see if improvements are possible. Of course there can be good reasons for a low ratio, such as having to use lower efficacy lamps or less efficient luminaire in order to achieve the required lighting result – but it is essential to check whether there is a scope for a more efficient alternative. Existing installations with an ILER of 0.5 or less certainly justify close inspection to identify options for converting the installation to use more efficient lighting equipment.

This process of comparing the installed load efficacy (ILE) with the target value for the Room Index and type of application can also be used to assess the efficiency of designs for new or replacement of general lighting installations. If, when doing so, the calculated ILE (lux/W/m^2) is less than the target value then it is advisable to ascertain the reasons. It may be that the requirements dictate a type of luminaire that is not as efficient as the best, or the surface reflectance are less than the normal maxima, or the environment is dirty, etc. Whatever the reasons, they should be checked to see if a more efficient solution is possible.

Solved Example:

Determine the cooling load of a commercial building for the following given data:

Outdoor conditions : DBT = 35°C, WBT = 25°C, Humidity = 18 g of water / kg of dry air
Desired indoor conditions : DBT = 25.6°C, RH = 50 %, Humidity = 10 g of water / kg of dry air
Total area of wall = 40 m^2, Total area of window = 20 m^2
U – Factor (Wall) = 0.33 W / m^{2}K
U – Factor (Roof) = 0.323 W / m^{2}K
U – factor [fixed windows with aluminium frames and a thermal break] = 3.56 W /m^{2}K

Other data:

- 15 m x 25 m roof constructed of 100 mm concrete with 90 mm insulation & steel decking.
- CLTD at 17:00 hr : Details : Wall = 12°C; Roof = 44°C; Glass Window = 7°C
- SCL at 17: 00 hr : Details : Glass Window = 605 W/ m^2
- Shading coefficient of Window = 0.74
- Space is occupied from 8:00 to 17:00 hr by 25 people doing moderately active work.
- Sensible heat gain / person = 75 W; Latent heat gain / person = 55 W; CLF for people = 0.9
- Fluorescent light in space = 21.5 W/m^2; CLF for lighting = 0.9
- Ballast factor details = 1.2 for fluorescent lights & 1.0 for incandescent lights
- Computers and office equipment in space produces 5.4 W/m^2 of sensible heat
- One coffee maker produces 1050 W of sensible heat and 450 W of latent heat
- Air changes / hr of infiltration = 0.3
- Height of building = 3.6 m

Solution:

I. External Heat Gain

(i) Conduction heat gain through the wall = U – factor x net area of wall x CLTD
= 0.33 x (40-20) x 12] = 79.2 W

(ii) Conduction heat gain through the roof = U – factor x net area of roof x CLTD
= 0.323 x (15 x 25) x 44
= 5 329.5 W

(iii) Conduction heat gain through the windows

= U-factor x net area of windows x CLTD

= (3.56 x 20 x 7) = 498.4 W

(iv) Solar radiation through glass = Surface area x Shading coefficient x SCL

= (20 x 0.74 x 605) = 8954 W

II. Internal Heat Gain

(i) Heat gain from people = Sensible heat gain + Latent heat gain

Sensible heat gain = (No. of people x Sensible heat gain / person x CLF)

= (25 x 75 x 0.9) = 1687.5 W

Latent heat gain = No. of people x Latent heat gain / person

= (25 x 55) = 1375 W

Therefore, Heat gain from people = (1687.5 + 1375) = 3062.5 W

(ii) Heat gain from lighting = (Energy input x Ballast factor x CLF)

Energy input = (Amount of lighting in space/unit area) x Floor area

= 21.5 x (15 x 25) = 8062.5 W

Therefore, heat gain from lighting = (8062.5 x 1.2 x 0.9) = 8707.5 W

(iii) Heat generated by equipment:

Sensible heat generated by coffee maker = 1050 W

Latent heat generated by coffee maker = 450 W

Sensible heat gain by computers and office equipment = 5.4 x 375 = 2025 W

Therefore, Heat generated by equipment = 3525 W

(iv) Heat gain through air infiltration = (Sensible heat gain + Latent heat gain)

Sensible heat gain = (1210 x airflow x ΔT)

Airflow = (Volume of space x air change rate) / 3600

= {(15 x 25 x 3.6) x 0.3} / 3600

= 0.1125 m^3 / s

Therefore, sensible heat gain = 1210 x 0.1125 x (35 – 25.6) =1279.58 W

Latent heat gain = 3010 x 0.1125 x (18 – 10) = 2709 W

No.	Space Load Components	Sensible Heat Load(W)	Latent Heat Load(W)
1	Conduction through exterior wall	158.4	----
2	Conduction through roof	5329.5	----
3	Conduction through windows	498.4	----
4	Solar radiation through windows	8954	----
5	Heat gained from people	1687.5	1375
6	Heat gained from lighting	8 707.5	----
7	Heat gained from equipment	3 075	450
8	Heat gained by air infiltration	1 279.58	2709
Total space cooling load		**29689.88**	**4534**
Total Cooling Load = **34,223.88 W**			

<table>
<tr><th colspan="2">QUESTIONS</th></tr>
<tr><td>S-1</td><td>Define average annual hourly EPI?</td></tr>
<tr><td>S-2</td><td>Define EPI?</td></tr>
<tr><td>S-3</td><td>Calculate the heat loss due to conduction for a wall with the following data:
Area of the wall : 9.3 m^2, U – factor : 0.284 $W/m^2\ ^oK$, Indoor dry bulb temperature : 21.1 ^oC, Out door dry bulb temperature : 30 ^oC.</td></tr>
<tr><td>S-4</td><td>Define ILER and its significance.</td></tr>
<tr><td>S-5</td><td>Define room index? For a room of length 10 m and width 20 m, calculate room index?</td></tr>
<tr><td>S-6</td><td>What are the thermal comfort standards for temperature and humidity as per NBC?</td></tr>
<tr><td>S-7</td><td>The built-up area of the building is 1500 m^2 and lighting load is 25 kW. Find out the lighting power density (LPD) of the building?</td></tr>
<tr><td>S-8</td><td>The annual energy consumption for a building with 8500 m^2 of built-up area is 32,00,000 kWh. Calculate the EPI?</td></tr>
<tr><td>S-9</td><td>The wet bulb temperature of the return air from an AHU is 29 ^0C. Find out the enthalpy from psychrometric chart.</td></tr>
<tr><td>S-10</td><td>Why water cooled refrigeration system is more efficient than air cooled?</td></tr>
<tr><td>L-1</td><td>In a performance monitoring of Air Handling Unit (AHU) in a building energy audit, the following observations were made.
(i) air coil filter area = 3.05 m^2 (ii) air velocity = 1.5 m/s (iii) Return air dry and wet bulb temperatures are 22.6 and 17.9 respectively (iv) Supply air dry and wet bulb temperatures are 15.7 and 15.2 respectively.
Calculate the actual TR load of AHU</td></tr>
<tr><td>L-2</td><td>The SHR for a given space is 0.9. If the latent heat gain is 1900 W, calculate the sensible heat gain.</td></tr>
<tr><td>N-1</td><td>Compare the performance of a centrifugal chiller with vapour absorption chiller using the data given below
<table>
<tr><th>Parameter</th><th>Centrifugal chiller</th><th>VAM</th></tr>
<tr><td>Chilled water flow (m^3/h)</td><td>189</td><td>180</td></tr>
<tr><td>Condenser water flow (m^3/h)</td><td>238</td><td>340</td></tr>
<tr><td>Chiller inlet temp (°C)</td><td>13.0</td><td>14.6</td></tr>
<tr><td>Condenser water inlet temp (°C)</td><td>27.1</td><td>33.5</td></tr>
<tr><td>Chiller outlet temp (°C)</td><td>7.7</td><td>9.0</td></tr>
<tr><td>Condense water outlet temp (°C)</td><td>35.7</td><td>39.1</td></tr>
<tr><td>Comp. power consumption (kW)</td><td>190</td><td>-</td></tr>
<tr><td>Steam consumption (kg/h)</td><td>-</td><td>1570</td></tr>
<tr><td>Chilled water pump (kW)</td><td>28</td><td>28</td></tr>
<tr><td>Condenser water pump (kW)</td><td>22</td><td>33</td></tr>
<tr><td>Cooling tower fan (kW)</td><td>6.0</td><td>15</td></tr>
</table>
i) Evaluate the tonnes of refrigeration (TR) of both the systems?
ii) Compare the auxiliary power consumption of both the chillers. What do you observe from the data and give reasons?</td></tr>
</table>

REFERENCES

1. Building Energy Audit Reports of National Productivity Council
2. TRANE Air-conditioning Clinic – Cooling and Heating Load Estimation
3. Energy Conservation Building Code – User Guide and Tip sheet, BEE, GOI

Chapter 15

Performance Assessment of Textile Industry

Learning Objectives

In this chapter you will learn about

- Textile manufacturing process
- Monitoring of energy consumption
- Performance assessment of process equipment and utilities

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Understand the process of textile manufacturing**
- ✓ **Monitor the energy consumption of process equipments and utilities**
- ✓ **Estimate the efficiency of stenters/dryers**
- ✓ **Assess the performance of dyeing machines**
- ✓ **Evaluate the performance of thermic fluid heater**

15. ENERGY PERFORMANCE ASSESSMENT OF TEXTILE INDUSTRY

15.1 Introduction

The textile industry in India is one of the largest process industries in terms of its production and employment. Modern textile units are energy intensive requiring both thermal and electrical energy for their operation. The energy cost accounts for about 15 – 20% of the manufacturing cost. A typical energy consumption pattern of a composite textile mill is given in Figure 15.1.

The textile industries with energy consumption equivalent to 3000 MTOE per year and above have been brought under designated consumer category as per Energy Conservation Act in India.

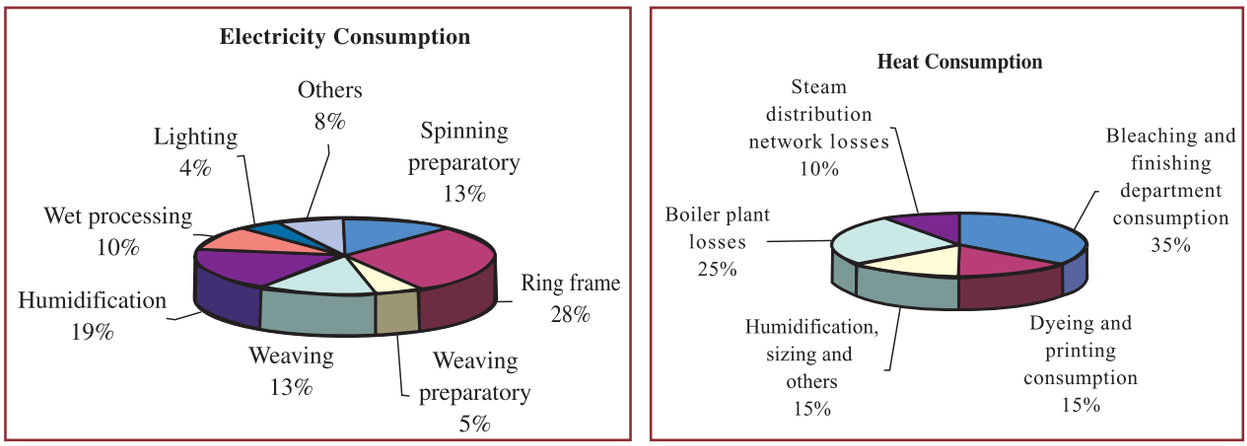


Figure 15.1 Energy Consumption of a Composite Textile Mill

15.2 Textile Manufacturing Process

The four major stages of processing is follows as (Figure 15.2)

- Spinning and weaving process (A)
- Wet Process (B)

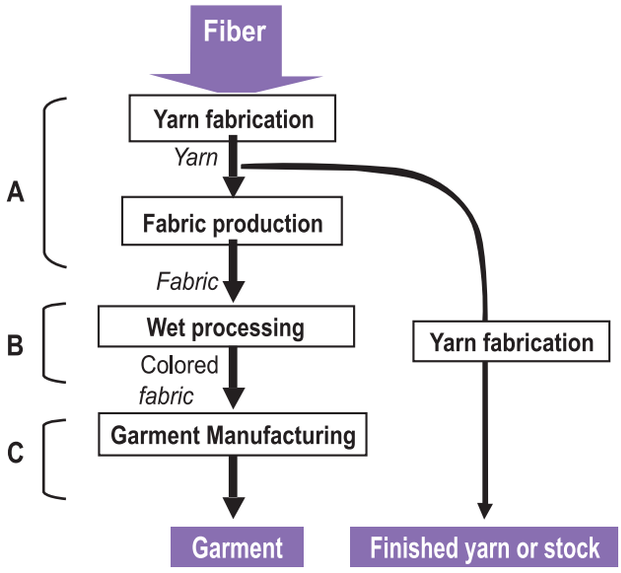


Figure 15.2 Textile manufacturing processes

Spinning and weaving

a) Yarn Fabrication

Yarn fabrication is the process, which converts raw fiber into yarn or thread (figure 15.3). The fibers are prepared and then drawn out and twisted to form the yarn, which is then wound onto a bobbin or cone. The yarn fabrication is entirely dry, although some yarns may be dyed and finished as a final customer product

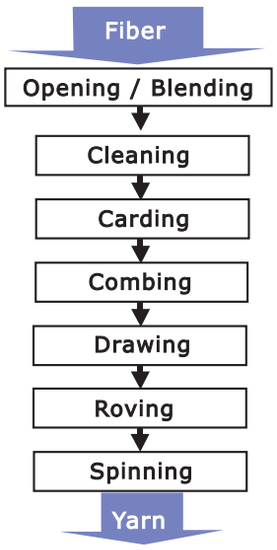


Figure 15.3 Main steps in yarn production of cotton and wool (Natural Fibers)

Opening / Blending: Raw materials (cotton and synthetic) are received in compact bales. The traveling type machine removes and separates the compressed fibers from each bale and open into

the smallest flock gently. More bales can be processed at a time by this type and thus have long term blending.

Cleaning: Cotton is passed through a series of machines (beaters) where opening and cleaning takes place simultaneously. Removes strange matters such as plant parts, seed hulls, dirt, etc, from the fibers. Speed of beater and clearance setting may be varied to achieve the desired level of cleaning.

Carding: The fibers are formed into a mat by passing through two press rollers. This mat is continuously fed into the card. The sheet of carded fibers is removed through a funnel into a loose rope like strand called a sliver.

Combing: Processing cotton or wool stock through a series of needles/combs, to remove short fibers and strange matters and to form a sliver of maximum possible evenness.

Drawing: The drawing frame contains number of strands of card sliver and passes them through a set of rollers that rotate at successively faster speed. These rollers combine fibers into a single and more uniform strand. They draw out and lengthen fibers (5-6 times).

Roving: The drawn frame sliver is fed into the drafting arrangement, which attenuate the sliver. The yarn is stretched and slightly twisted to strengthen the strand (8 times the length and 1/8 the diameter).

Spinning: The rove fed into the drafting arrangement is further attenuated and twisted to the desired size to spin into the final yarn. The twist is generated by the spindle, which rotates at a higher speed. Each revolution of spindle imports one turn of twist to the strand. Finally the yarn (twisted fiber strand) is wound on bobbin mounted on spindle with the help of traveler.

It uses four types of technologies: Ring spinning, rotor spinning, air jet spinning and friction spinning. Ring spinning is the most used in India due its wide adaptability for spinning different types of yarn. Rotor spinning is also widely used.

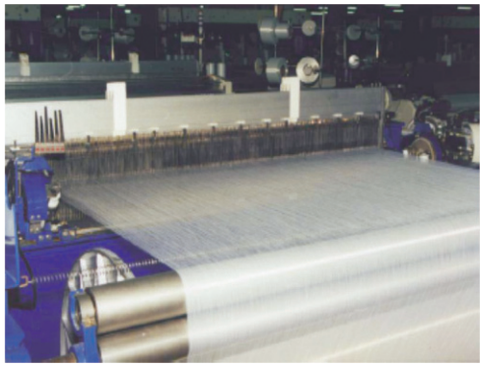
b) Fabric Production

Fabric production, the second step, involves weaving / knitting

Weaving :

Weaving is the most common method used for producing fabrics. The process is carried out of two sets of threads, which interlaces lengthwise yarns (warp yarns) with widthwise ones (weft or filling yarns).

To prevent the warp yarns from braking during weaving, the warp threads are coated with a size at 80 – 90 °C before weaving, to increase their tensile strength and smoothness. Natural starches are the most commonly used sizes, although compounds such as polyvinyl alcohol (PVA), resins, alkali-soluble cellulose derivatives, and gelatin glue have been used. The sizing compound is dried at 100-130 °C on the threads and remains a part of the cloth until it is removed in the subsequent processes.



Knitting :

In knitting fabric is formed through interlocking series of yarn loops by using hooked needles. Rows of stitches are formed so that each row hangs on the row behind it, usually using sophisticated, high-speed machinery.

Wet Processing

The fabric produced from the weaving or knitting is in rough condition, contains impurities and is often termed 'grey' fabric. Fabric processing (wet processing) is done to improve the appearance and serviceability of the fabric. The main operations carried out in this step include pre-treatment, dyeing, printing and finishing (figures 15.4 and 15.5). The wet processing, consumes huge quantities of water, chemicals and thermal energy.

Figure 15.4 Wet processing of knitted cotton fabrics

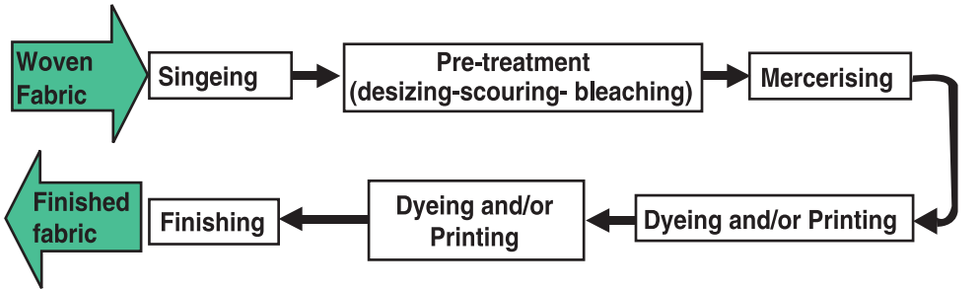


Figure 15.5 Wet processing of woven cotton fabrics

Summary of the main wet processing operations in textiles manufacturing	
Process	**Purpose and Conditions**
Singeing	Direct or indirect flames are used to remove fuzzy fibers, followed by quenching.
Desizing	The sizing ingredients are removed from the grey fabric by dissolving them. Desize formulation depends on the nature of the sizing agents, i.e. enzymatic or oxidative treatments are used for starch sized fabrics, whilst acrylate sizes can be removed by hot water (80-90^{o}C). PVA can be removed by using hot water in the presence or absence of peroxygens.
Scouring	The scouring process is carried out to remove impurities such as wax, fatty acids, oils, etc, present in the fabric. Scouring is carried out in alkaline conditions (with sodium hydroxide) under high pressure and temperature (above 100^{o}C).
Bleaching	Bleaching is used to whiten fabrics and yarns. Different chemicals such as hypochlorites (room temperature), hydrogen peroxides (temperature 90-95^{o}C), etc, are used as bleaching agents. The process conditions during bleaching vary with the type of agent used. Once bleaching is complete, the bleaching agent must be completely removed, either by thorough washing or through the use of enzymes.
Mercerization	Mercerizing increases the tensile strength, luster and dye uptake of the cotton fabric or yarn, which is treated with cold sodium hydroxide solution (temperature 18^{o}C, time 20-40 sec). This causes swelling of the fiber, which results in an increase in the dye intake
Dyeing	Dyeing is employed to give an all-over shade to the fabric. It basically involves diffusion of dye molecules into the textile fabric, which imparts the required color. Dye bath formulation and dyeing conditions depends on the dye stuff classes used. Batch and Continuous techniques are available for dyeing the textile material. Jigger, Winch, Padding, Mangle, and Jet dyeing are some of the important dyeing machines used.
Printing	Printing is a process by which colored patterns are produced on the fabric. The color is applied to the prepared fabric on specific areas and then treated with steam, heat or chemicals to fix the color to achieve a planned design. Pigment or wet or discharge printing can also be used. Final washing of the fabric is carried out to remove excess paste and leave a uniform color.

Finishing

This stage includes the final operations necessary for making the textile presentable and attractive. It imparts the final aesthetic, chemical and mechanical properties to the fabric as per the end use requirements to improve appearance, texture or quality. The finishing operations include:

Drying: Drying removes the moisture from the fabric using drying machines such as stenter / drying cylinders.

Providing Dimensional Stability: This is one of the most important finishing operations. The fabric, which is in a distorted condition, is brought to the required dimensions of width and length in a

process called heat setting at around 140 - 150 ^{o}C. The heating medium used is thermic fluid/steam.

Calendering: A kind of glossy skin is formed on the fabric surface during calendering. The damp fabric is pressed hard against a hot, polished metal surface until it dries.

Softening: After calendering, the fabric becomes a little stiff. Breaking down this stiffness is called softening. The fabric is led through the softening machine so that it touches the studded rollers lightly and drags them around. In this way, the surface of the fabric is lightly disturbed making it much softer.

Depending upon the type of fabric to be processed and the final product, any or all of the above processing operations can be carried out.

15.3. Monitoring of Energy Consumption in Textile Production Process

The objective of the assessment is to find out the specific energy consumption of both electrical and thermal per kg of yarn or fabric. The main processing areas in textile manufacturing that consume more energy are Spinning and Weaving, Wet Processing, and Garment Manufacturing. In addition, processes not directly related to production such as compressors, humidification systems etc. also consume significant energy.

Assessment of electrical energy consumption in the process

a) Spinning and weaving processes

Determine the processes in spinning and weaving where the largest amount of electricity is used and the data is to be tabulated as per table 15.1.

Table 15.1. Overview of electricity use in Spinning and weaving processing

Process/equipment	Specific energy consumption

b) Wet processing

Determine the processes in wet processing where the largest amount of electricity is used and the data is to be tabulated as per table 15.2.

Table 15.2. Overview of electricity use in wet processing

Process/equipment	Specific energy consumption

Assessment of electrical energy consumption in utilities (other than process)

Electrical energy is used up in many areas other than process in the textile plant, including humidification, compressed air systems, lighting systems, electrical distribution network (as losses) and other utilities. Each major utility has to be assessed for energy consumption using online energy meters if available or with the portable power analyzer and tabulated as in table 15.3.

Table 15.3. Electrical energy use in other activities

Process	Specific energy consumption
Humidification	
Compressed air systems	
Lighting	
Other utilities	

Assessment of overall electrical energy use

After reviewing the main production areas as well as the other company activities, this step is intended to analyse the overall energy use and to calculate associated costs and ratios as shown in table 15.4.

By comparing the current performance with the previous years or the industries average (benchmarking) it will be possible to identify energy saving opportunities. Since energy use may vary according to capacity utilization, the use of relative indicators, such as (kWh / kg of product) and (total energy costs / overall operating expense) is encouraged. This step will also help to continue with the further development of potential energy-saving changes.

Table 15.4. Overview of total electricity use in textile production

Area or Activity	(a) Total electricity use (kWh)	(b) Unit Cost (Rs./kWh)	(c) = (a) x (b) Total electricity costs
Spinning & weaving			
Wet processing			
Other activities			
Total electrical energy use			

Assessment of overall thermal energy use

Besides electricity, thermal energy carriers such as steam and thermic fluid generated through coal, gas, oil, wood etc., are also used in other processes and activities. The thermal energy is predominantly used in wet processing activities. Since fuels have different calorific values thermal energy used is

expressed in Metric Tonnes of Oil Equivalent (MTOE). The format for assessment of overall thermal energy use is given in table 15.5.

Table 15.5. Total thermal energy use

Energy carrier	Total fuel consumption (a)	Calorific value kcal/unit (b)	Total energy use (MTOE) (c) = (b) / 10^7	Cost per unit (d)	Total costs (e) = (a) x (d)
Fuel oil, kg					
Coal, kg					
Natural gas, Sm^3					
Charcoal, kg					
Wood, kg					
Others					
Total thermal energy use					

Assessment of overall energy use

The overall energy use can be calculated from the electricity bills and fuel bills for one year period and expressed in MTOE as shown in table 15.6.

Table 15.6. Total overall energy in textile production

Overall energy use	Total energy use (MTOE)
Electrical energy (Grid + Captive generation), (kWh x 860 / 10^7)	
Thermal energy, MTOE	
Total overall energy use	

Benchmarking / Energy Performance Index

The generated data can be further used for benchmarking as shown in table 15.7. In an internal benchmarking, the current performance can be compared with past values in order to observe the improvements. External benchmarking will require comparison with industry average or competitors. The benchmarking parameters commonly used are kWh/kg and kcal/kg of yarn or fabric.

Table 15.7. Benchmarking / Performance Index

Specific energy consumption	Benchmark	Actual
Electrical – kWh/kg (Grid + Captive generation)		
Thermal energy, kcal/kg		

15.4 Performance Monitoring of Process Equipment

The major energy consuming equipment in textile processes are ring frames, carding, winding and blow room machinery. Since the ring frame is single largest consumer, the performance monitoring is limited to ring frame. Similar approach can be adopted for other equipment. In the wet process the energy intensive equipment are stenters and dyeing machines, performance evaluations of which are given in this chapter.

Performance Evaluation of Ring Frame

The energy consumption depends on yarn count, mix ratio of cotton and polyester, spindle speed, ring diameter, weight of traveler, package diameter, lift, spindle gauge, spindle wharve diameter, type of spindle tape used, front roller speed, maintenance practices of ring frames, type of pneumafil system, motor drive system, etc. The spinning energy performance is reported in terms of kWh consumption per kg of yarn produced at an average reference count. Also there is a convention of reporting specific energy consumption as kWh consumption per spindle shift. Both the figures are bench marked against best practice values and energy saving opportunities implemented accordingly.

Motor Performance Monitoring:

Since the spinning section comprises of large number of motors, load analysis of motors has to be carried out and potential for shuffling, operating in star mode, retrofitting of soft starters energy saver, replacement of motors etc need to be analysed. The procedures described in Chapter-5 may be adopted.

Performance Evaluation of Stenter –Hot Air Dryer

The stenter is a hot air dryer, with the fabric passing through on a chain drive, held in place by either clips or pins. The stenter has many chambers and in each chamber, the air is blown through the radiators in which steam or thermic fluid is passed. The heated air is circulated above and below the fabric and is exhausted to the atmosphere by exhaust fans. Direct firing of natural gas is also practiced in stenters. Apart from drying, the stenter is a versatile piece of equipment used for pulling the fabric to width, chemical finishing, heat setting and curing. Refer figure 15.6 (a & b) for the schematic diagram.

Performance Evaluation Methodology

Efficiency estimation of Stenters/dryers can be done by measurement of moisture content in material before and after the dryer to estimate total moisture removal from the substance. The energy required to drive out this moisture is termed as useful energy spent in the dryer. By measuring the total input heat energy to the dryer, the dryer efficiency is estimated.

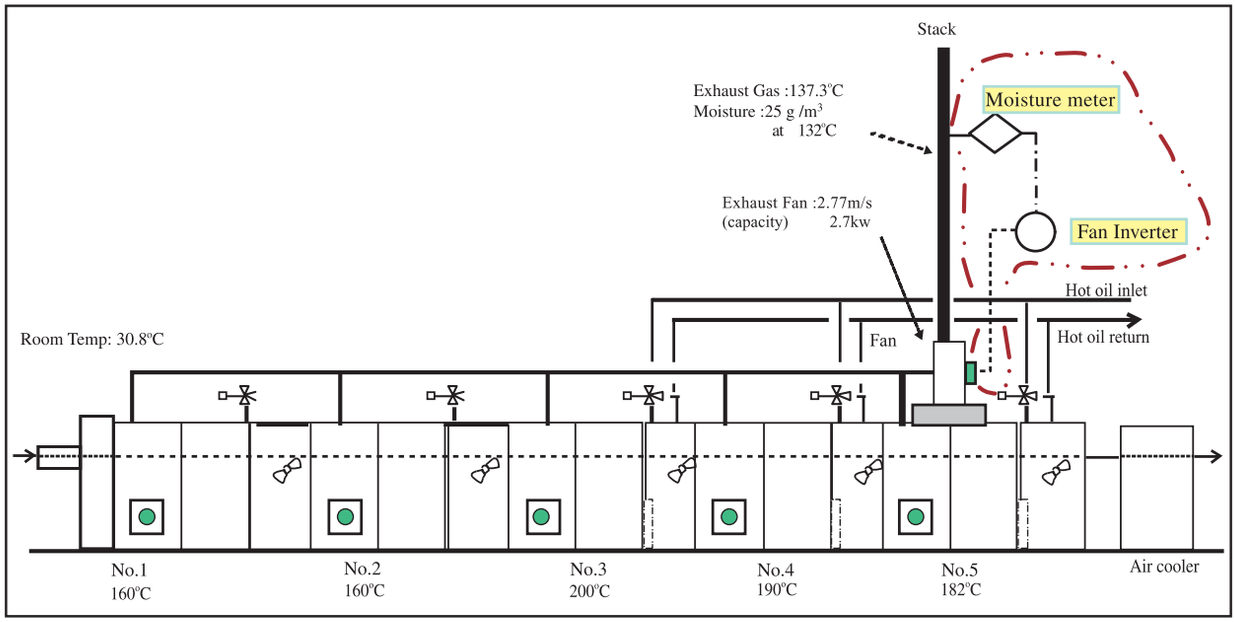


Figure 15.6 (a) Schematic of a Stenter

Figure 15.6 (b) Typical Stenter

Parameters to be measured

1. Measure moisture content of material at inlet to dryer
2. Measure moisture content of material at outlet of dryer
3. Measure mass flow rate of dried material.
4. Measure input thermal energy to dryer which could be in the form of (i) hot air input (ii) Steam heating or (iii) electrical heating through various direct measurements or indirectly from quantity of fuel fired and combustion efficiency assessments for direct fuel fired dryers.

$$Dryer\ efficiency, \% = \frac{W \times (m_{in} - m_{out}) \times \{(T_{out} - T_{in}) + 540\}}{Q_{in}} \times 100$$

Where,

W - Weight of the material output of the dryer on bone dry basis, kg/hr

m_{in} and m_{out} - Moisture content in feed and output material of the dryer, respectively, expressed as kg moisture/kg bone dry material

T_{in} and T_{out} - Temperature of the material being dried at the dryer inlet and outlet, respectively, ^{0}C

Q_{in} - Thermal energy input rate to dryer, kcal/hour

Example 15.1

The throughput of a textile stenter is 1000 kg/hr. The inlet moisture content of the cloth is 50% and the outlet moisture is 5%. The inlet and outlet temperature of the cloth is 28 ^{0}C and 80 ^{0}C. The stenter is supplied with steam at an enthalpy of 660 kcal/kg, the condensate leaves at 90 ^{0}C. The flow rate of steam is 1000 kg/hr. Estimate the overall thermal efficiency of the dryer. The latent heat of evaporation of water is 540 kcal/kg.

1000 kg/hr of wet cloth contains

1000 x 0.50 kg water = 500 kg/hr of moisture

and 1000 x (1-0.50) = 500 kg/hr bone dry cloth.

As the final product contains 5% moisture, the moisture in the product is (500 x 0.05)/0.95 = 26.3 kg

Inlet moisture = 500/500 = 1 kg of moisture / kg of bone dry material

Outlet moisture = 26.3/500 = 0.053 kg of moisture / kg of bone dry material

$$Dryer\ efficiency, \% = \frac{W \times (m_{in} - m_{out}) \times \{(T_{out} - T_{in}) + 540\}}{Q_{in}} \times 100$$

$$\text{Dryer efficiency}, \% = \frac{500 \times (1 - 0.053) \times \{(80 - 28) + 540\}}{1000 \times (660 - 90)} \times 100$$

$$\text{Dryer efficiency} = 49\%$$

Performance Evaluation of Dyeing Machines

The different types of machines used in the aqueous dyeing of textile fabrics are open width Jigger dyeing (figure 15.7), Jet dyeing, rope form Winch dyeing etc. These are operated at atmospheric pressure or higher pressure and temperatures below or above 100°C depending on the substrate being dyed. The major energy usage in dyeing is thermal energy in the form of steam.

The bath (liquor) ratio may vary from 1:4, 1:6 to 1:10 or higher (i.e.1 kg of textile material to 4, 6 to 10 litres of water on weight ratio) based on the dyeing types (batch / continuous). The dye colour and chemicals are mixed in the water heated by steam.

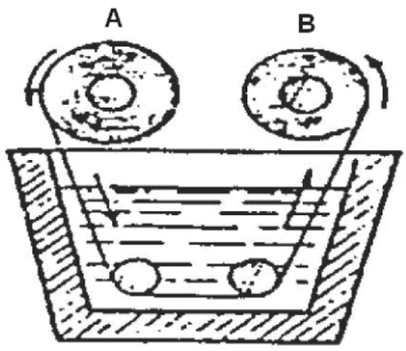


Figure 15.7 Jigger Dyeing

Table 15.8. Effect of material – to – Liquor ratio on consumption of steam

Liquor Ratio	Steam kg per kg of fabric	
	Remazol Dyes (40°C)	Disperse Dyes (130°C)
1:4	1.00	1.80
1:6	1.60	2.65
1:8	2.00	3.60
1:10	2.60	4.40
1:12	---	5.30
1:20	5.35	---

Table 15.9. Energy requirement for dyeing cotton fabrics

Process	Liquor Ratio	Temperature (°C)	Energy MJ/kg		
			Dyeing	Washing	Total
Winch, hot reactive	1:2 1:3	80 11.5	8.1 11.5	7.8 11.3	15.9 22.8
Winch, direct	1:2 1:3	95 95	9.7 13.1	0.4 0.4	10.1 13.5
Winch, cold reactive	1:2 1:3	30 30	2.3 3.0	7.8 11.3	10.1 14.3
Closed Jig, direct	1:2 1:3	95 95	1.8 2.3	0.0 0.0	1.8 2.3
Closed Jig, cold reactive	1:2 1:3	30 30	0.4 0.7	3.9 3.9	4.3 4.6
Closed Jig, Hot reactive	1:2 1:3	80 80	1.8 2.0	3.9 3.9	5.7 5.9

Energy Balance Calculation for dyeing machines

The approach to calculate energy balance is more or less similar to all dyeing machines. Dyeing requires steam to heat the liquor to the requisite temperature and maintain the temperature for a specific period of time.

Energy input is calculated in the following ways:

a) Steam flow rate can be calculated by collecting the quantity of steam condensate for a specified time period.

b) Enthalpy of steam can be drawn from the pressure of steam using steam table/Mollier chart.

Heat Load (at input) = Ms x total enthalpy of steam

Energy output is calculated in the following ways:

a) The heat absorbed by the fabric and vessel.
b) The heat taken by the liquor.
c) The heat through radiation and convection by machine surface
d) Heat in the vapours

Heat Load (at output) = (b) + {(a) + (c) + (d)}* = (M_l Cp_l Δt_l) + {10% of (b)}

Where, Ms – Mass flow rate of steam
M_l – Mass of liquor
Cp_l - Specific heat capacity of liquor
T – Temperature difference between initial and final liquor temperatures

* Since the values are insignificant and tedious to calculate, it will be reasonable to assume that fabric and other losses account for additional 10% of the steam input.

Specific consumption of steam in dyeing is calculated as kg of steam per kg of cloth.

Example 15.2. 100 kg of fabric is to be dyed in a jigger. The dye liquor is heated from 30 to 80 °C. Calculate the quantity of steam required at 2 kg/cm^2 (g) and specific steam consumption if the liquor ratio is 1:6.

Using heat balance equation,
Heat energy input = Heat energy output

i.e., Ms x Enthalpy of steam = (M_l Cp_l Δt_l) + {10% of M_l Cp_l Δt_l}

Enthalpy of steam at 2 kg/cm^2 (g) = 650 kcal/kg

Given Liquor ratio = 1:6, which implies to dye 100 kg of fabric, 600 litres of water is required

$$\text{Therefore, Quantity of steam (Ms)} = \frac{\{[600 \times 1 \times (80\text{-}30)] + 0.1 \times [600 \times 1 \times (80\text{-}30)]\}}{650}$$

$$= 50.76 \text{ kg}$$

Specific steam consumption = 50.76 / 100 = 0.51 kg steam / kg cloth

15.5 Performance Monitoring of Utilities

The major energy utilities of textile industry are:

i. Boilers and steam systems
ii. Thermic fluid heaters
iii. Compressed air systems
iv. Humidification systems

Boiler and Steam Systems

The steam boiler, as a supplier of thermal energy for heating purposes, is the heart of a dye house in wet processing operations. Different fuels are used to operate the steam boilers viz., wood, coal, oil and gaseous fuels based on the cost and availability, calorific value, ease of combustion and flue gas characteristic.

Refer to chapter-1 of this book for **Performance evaluation of boiler** (Direct & **Indirect method of efficiency testing**) and chapter-3 of book - 2 Steam systems.

Thermic Fluid Heaters

The thermic fluid heaters provide thermal energy for high temperature operations such as drying of fabric in stenter after dyeing operation in wet processing. The drying and finishing of textiles are highly energy intensive processes since it involves the use of very hot air (140-200^{o}C) to either remove chemically held water from the textiles by evaporation or physically /chemically set the textile materials.

Employing petroleum - based fluids as the heat transfer medium, these heaters provide constantly maintainable temperatures for the user equipment.

The modern oil fired thermic fluid heater (refer figure 15.8) consists of a double coil, three pass construction and fitted with modulated pressure jet system. The thermic fluid, which acts as a heat carrier, is heated up in the heater and circulated through the user equipment. There it transfers heat for the process through a heat exchanger and the fluid is then returned to the heater. The flow of thermic fluid at the user end is controlled by a pneumatically operated control valve, based on the operating temperature. The heater operates on low or high fire depending on the return oil temperature, which varies with the system load.

The advantages of these heaters are:

- Closed cycle operation with minimum losses as compared to steam boilers.
- Non-Pressurized system operation even for temperatures around 250 ^{0}C as against 40 kg/cm^{2} steam pressure requirement in a similar steam system.
- Automatic control settings, which offer operational flexibility.
- Good thermal efficiencies as losses due to blow down, condensate drain and flash steam do not exist in a thermic fluid heater system.

Performance Evaluation

Thermic fluid heater efficiency can be tested by the following methods:

1. **The Direct Method:** Where the energy gain of the working thermic fluid is compared with the energy content of the heater fuel.

2. **The Indirect Method:** Where the efficiency is the difference between the losses and the energy input. It is also called as heat loss method.

The Direct Method of Testing: This is also known as 'input-output method' due to the fact that it needs only the useful output in thermic fluid and the heat input (i.e. fuel) for evaluating the efficiency. This efficiency can be evaluated using the formula

$$\text{Thermic fluid heater efficiency} = \frac{\text{Heat output}}{\text{Heat input}} \times 100$$

Where,

Heat Output (kcal/h) = mass flow rate of thermic fluid (kg/h) x specific heat of fluid (kcal/kg°C) x temperature difference of fluid (°C)

Heat Input (kcal/h) = Fuel firing rate (kg/h) x Gross calorific value of fuel (kcal/kg)

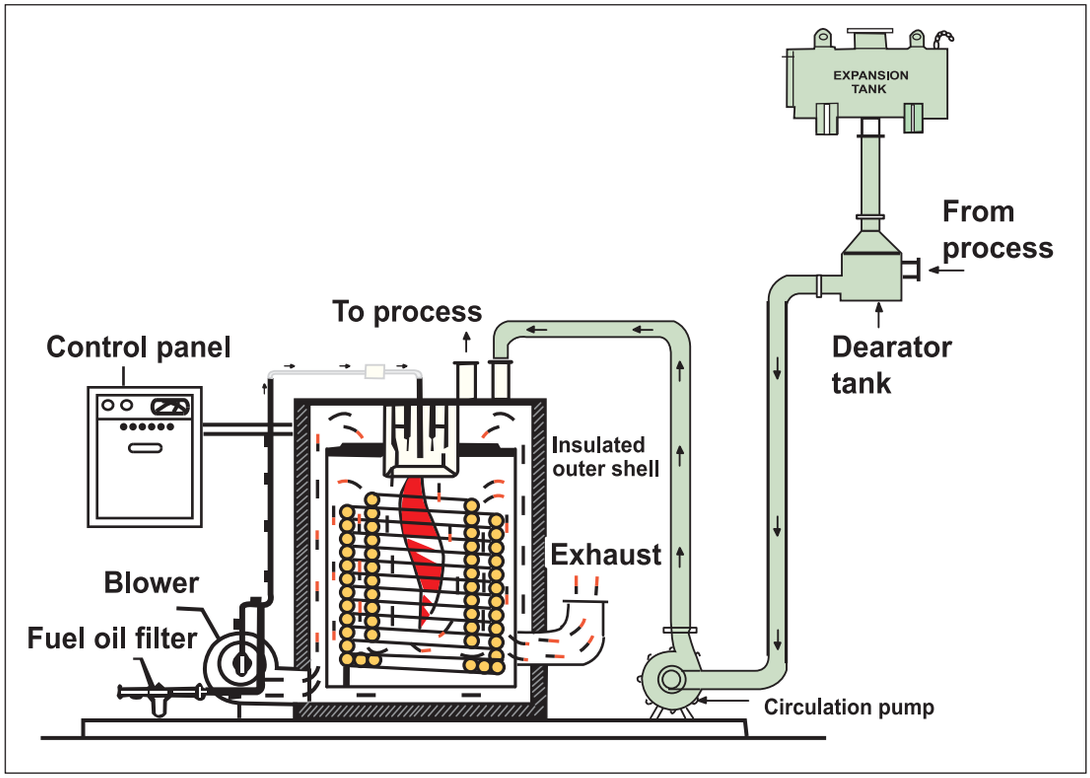


Figure 15.8 A typical configuration of Thermic Fluid Heater
(Energy Machine India)

Example 15.3. Test Data and Calculation

Heat output data:

Flow rate of thermic fluid : 68 m^3/h
Inlet temperature of thermic fluid : 200 ^oC
Outlet temperature of thermic fluid : 212 ^oC
Specific heat of thermic fluid : 2.223 kJ/kg ^oC
Density of the thermic fluid : 826 kg/m^3

Heat input data:

Quantity of fuel oil consumed (Input) : 46 kg/h
GCV of fuel oil : 10000 kcal/kg

Calculation:

Heat output (thermic fluid) = 68 x 826 x (2.223/4.187) x (212-200) = 357902.5 kcal/h

Heat input (fuel) = 46 x 10000 = 460000 kcal/h

Thermic fluid heater efficiency = (357902.5 / 460000) x 100 = 77.8%

(**Note:** The indirect method for efficiency testing of thermic fluid heater is similar to that for boiler testing; refer to chapter -1 Boiler, for Indirect Method)

Compressed Air Systems

The intensity of compressed air use is high in spinning section due to automated machines in spinning section. Mostly reciprocating/screw compressors are used. For the performance evaluation of compressed air systems, refer to chapter-8 of this book and chapter-3 of book-3.

Humidification Systems

Need for Humidification: The relative humidity and temperature level needs to be maintained within a prescribed tolerance limits at all steps of textile processing so as to avoid the problems of yarn breakage in dry and brittle condition and also to minimize the build up of static charge so as to reduce dust and fibre fly (fluff). For this purpose, spray air washers using spray water as the medium for adiabatic cooling of air (by direct evaporation of water into the air stream thereby reducing the air's dry bulb temperature and raising its humidity) are extensively used in humidification systems.

The system (figure 15.9) consist of a chamber containing multiple banks of spray headers with spray nozzles, a tank for collecting spray water as it falls and an 'eliminator section' with PVC blades having 3 or 4 bends for removing droplets of water from the air which is humidified after passing through the curtain of spray water, before discharge to the air ducts for distribution to the humidified areas.

Psychrometrics of humidification:

The room sensible heat load calculations are worked out as the sum-total of:

- Heat dissipation from the machines operating kW load in various departments.
- Lighting load
- Solar heat gain through insulated roof
- Occupancy load
- Heat gain from supply air fan motors

Note: For detailed heat load estimation, ASHRAE - HVAC handbook can be referred.

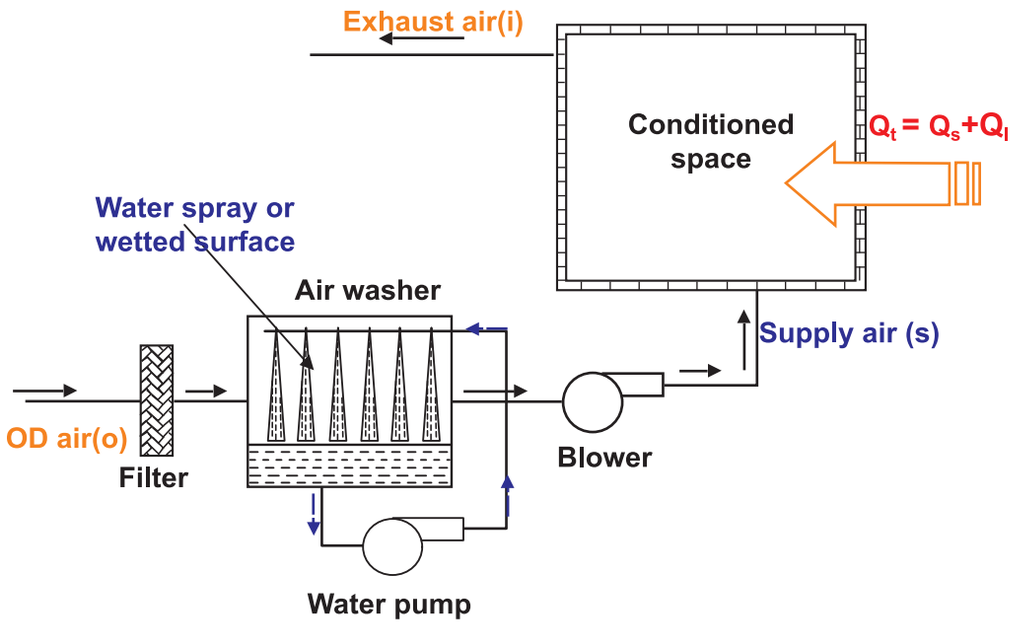


Figure 15.9 Typical Textile Humidification

Saturation efficiency

Higher saturation efficiency of air washer is defined as the ratio

$$\text{Saturation Efficiency (\%)} = \frac{\text{Entering Air DBT - Leaving Air DBT}}{\text{Entering Air DBT - Entering Air WBT}} \times 100$$

Improving the saturation efficiency reduces the temperature of air leaving the air washer thereby increasing the delta 'T' (temperature rise of saturated air due to heat pick up in the department) for a specific heat load and relative humidity condition in the department with consequent reduction in required supply air quantity and fan motor power consumption.

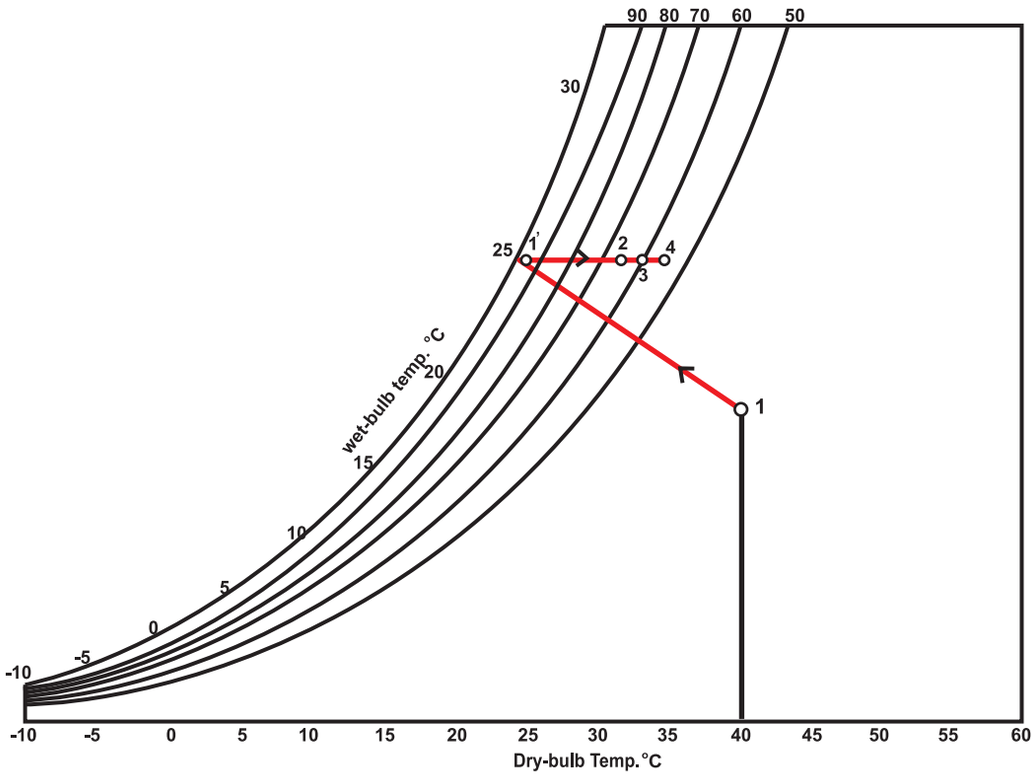
Example 15.4.

Temperature of humidified air leaving the air washer based on summer outside design dry bulb (40°C) and wet bulb temperatures (25°C) and design saturation efficiency (95%) is calculated as follows and plotted on the psychrometric chart (figure 15.10).

Leaving Air Temperature from Air - Washer

= Entering Air DBT - (Ent. Air DBT - Ent. Air WBT) ×% Saturation efficiency

Air temperature leaving the air washer = 40-(40-25) x 0.95 = 40-14.25 = 25.75°C.



PSYCHROMETRIC CHART TEMPERATURE IN ^{0}C

1 Entering air condition : 40 ^{0}C DB & 25 ^{0}C WB temperature to air washer
1' Air temperature leaving air washer 25.75 ^{0}C
2 Blow room : 65% RH & 31.9 ^{0}C
3 Spinning section : 60% RH & 33.2 ^{0}C
4 Carding, Combing, Drawing & Roving : 55% RH & 35 ^{0}C.

Figure 15.10 Mill Humidification Process

With change in ambient air wet bulb temperature, the air-temperature leaving the air washer will also vary along with a change in the department's inside dry bulb temperature, even though the inside design relative humidity condition remains constant.

The line of supply air temperature rise in the department due to heat gains is plotted horizontally, starting from the point of "leaving air temperature condition" after humidification in air washer, till it reaches the design relative humidity line for the corresponding department.

The required supply air quantity is calculated as follows:

$$Supply\ Air\ Quantity\ (m^3/h) = \frac{Room\ Sensible\ Heat\ Load\ (kCal/h)}{Specific\ heat\ of\ air\ (kCal/kg\ ^{O}C) \times Density\ of\ air\ (kg/m^3) \times Temperature\ rise\ of\ humidified\ air\ (^{O}C)}$$

Example 15.5. In a typical spinning mill humidification application, the sensible heat load of the spinning department is estimated as 10,03,000 kcal/h, temperature rise was 8°C and the density and specific gravity of the air were 1.2 kg /Nm^3 and 0.24 kcal/kg °C, respectively. Calculate the supply air quantity for the department.

Supply air quantity = (10,03,000 kcal/h) / (0.24 kcal/kg °C x 1.2 kg /Nm^3 x 8°C)

= 4,35,330 m^3/h.

Table 15.10. Humidification design parameters for a typical spinning mill with summer outside design conditions of 40 °C DBT & 25 °C WBT

S. No.	Department	Volume of space (m^3)	Machinery operating kW load	Inside Design Conditions		Estimated supply air flow rate (m^3/h)	Air changes per hour
				Relative humidity (%)	Inside DBT (°C) (at summer outside condition specified)		
1.	Blow Room	3150	96	65	31.9	35,834	20
2.	Carding, combing, draw frames and roving (preparatory areas)	11,300	422	55	45.0	1,67,000	15
3.	Ring Frame (Spinning)	13,200	1050	60	33.2	4,70,000	36

Performance Evaluation of Fans

The energy consumption in humidification systems is dominated by fans accounting for 90%. Hence any attempt to reduce energy consumption in humidification system depends on efficient operation of fans, optimizing number of air changes, reducing pressure drops in ducting and use of energy efficient fans and motors. For performance evaluation of fans, blowers and duct systems, refer to chapter-6 of this book.

Solved Example:

1) 125 kg of fabric is to be dyed in a jigger. The dye liquor is heated from 30 ^{0}C to 90 ^{0}C. Calculate steam (steam enthalpy 660 kcal/kg) requirement per batch and specific steam consumption (kg of steam per kg of cloth), if liquor ratio is 1:6.5; allowing 10% margin for losses.

2) In a textile mill, a thermic fluid heater of 15 lakh kcal/hr capacity is meeting process heat requirements.

The observed parameters of thermic fluid heater are:

Thermic fluid circulation rate	- 100 m^3/hr
Outlet temperature of fluid	- 270 ^{0}C
Return temperature of fluid	- 256 ^{0}C
Specific heat of fluid	- 0.55 kcal/kg ^{0}C
Density of fluid	- 830 kg/m^3
Present coal consumption	- 300 kg /hr
GCV of coal	- 3500 kcal/hr

i) What is the % loading of the thermic fluid heater?
ii) What is the existing thermal efficiency?
iii) The management is proposing to add a five chamber, (each chamber 1 Lakh kcal/hr duty) to the heater. Will it be able to take the load?

Solution:

1) Heat energy required / batch

= 125 kg x 6.5 liq.ratio x 1.10 (including margin) x (90-30)
= 53625 kcal/batch

Steam quantity needed / batch = 53625/660
= 81.25 kg

Specific steam consumption = 81.25/125 = 0.65 kg/kg

2) i) Heat duty = 100 m^3/hr x 830 kg/m^3 x 0.55 x 14
= 6,39,100 kcal/hr

% loading = 6,39,100/15,00,000
= 42.60 %

ii) Input energy = 300 kg/hr x 3500 kcal
= 10,50,000 kcal/hr

Efficiency of the heater = 6,39,100/10,50,000
= 60.86%

iii) New chamber heat duty = 5 x 1,00,000
= 5,00,000 kcal/hr

The total heat duty with new stenter loading = 6,39,100 + 5,00,000
= 11,39,100 kcal/hr

It is possible to comment new stenter since this is within 15 Lakh kcal/hr capacity of the TFH.

QUESTIONS	
S-1	In textile process what is meant by wet processing?
S-2	By what process wax, fatty acids and oils are removed from yarn/clothes in textile processing?
S-3	Name two chemicals used as a bleaching agent.
S-4	Name two kinds of machine used for dyeing.
S-5	200 kg of fabric is to be dyed in a jigger. The dye liquor is heated from 30 to 90 ^{o}C. The liquor ratio is 1:6. How many kcal of heat is required for heating the liquor. Specific heat of dye liquor is 1 kcal/kg ^{0}C.
S-6	By which process the tensile strength of cotton fabric is increased
S-7	List two advantages of using thermic fluid over steam
S-8	A textile industry operates two shifts a day. Though no production takes place during night shift, the thermic fluid pump is in operation to keep the thermic fluid in circulation. What energy saving option will you suggest as an energy auditor?
S-9	In a jet dyeing machine the condensate is discharged bypassing the trap. List two impact of this operation.
S-10	The dry bulb and wet bulb temperatures of air entering an air washer are 35 and 28 ^{0}C respectively. If the saturation efficiency is 90 %, calculate the air temperature leaving the air washer.
L-1	100 m^3 of hot dye liquor is discharged at 90 ^{0}C from the textile plant every day. How much kg of hot water can be obtained at 60 ^{0}C through waste heat recovery from dye liquor? The other data are as follows Specific heat of water and dye liquor : 1 kcal/kg ^{0}C Raw water temperature : 30 ^{0}C Desirable dye liquor discharge temperature : 40 ^{0}C. Density of dye liquor : 1000 g/lit
L-2	A textile industry consumes 3000 Tonnes of fire wood for its boilers and thermic fluid heaters and 12. 46 lac kWh per annum. Determine if this industry will be a designated consumer under the EC Act.

N-1

The following are the data collected for a thermic fluid heater using oil as the fuel. Find out the thermic fluid heater efficiency by indirect method. Assume heat loss due to radiation and convection as 1.5%

Fuel Analysis:

Carbon	84	%
Hydrogen	12.0	%
Nitrogen	0.5	%
Oxygen	1.5	%
Sulphur	1.5	%
Moisture	0.5	%
GCV	10700	kcal/kg

Ambient air properties

DBT	27.2	^{o}C
WBT	20.0	^{o}C
Abs. Humidity	0.012	kg / kg dry air

Thermic Fluid

Temperature return	220	^{o}C
Temperature Out	232	^{o}C

Flue Gas:

Oxygen (Furnace Out)	4.7	%
Carbon dioxide (Furnace Out)	12.2	%
Carbon monoxide	54	ppm
Temp. (Furnace Out)	325	^{o}C

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The following chapters:

Chapter 16 - Performance Assessment of Pulp and Paper Industry

Chapter 17 - Performance Assessment of Fertiliser Industry

are for reference and will not be a part of the examination syllabus until further notification.

Chapter 16

Performance Assessment of Pulp & Paper Industry

Learning Objectives

In this chapter you will learn about

- Pulp and paper manufacturing process
- Energy Interaction in pulp and paper manufacture
- Material and energy balance calculations for pulping digester and paper machine

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Understand the process of pulp and paper making**
- ✓ **Perform material and energy balance of digester and paper machine**
- ✓ **Estimate specific energy consumption**

16. ENERGY PERFORMANCE ASSESSMENT OF PULP & PAPER INDUSTRY

16.1 Introduction

Paper is produced from cellulose pulp fibers, derived from wood, rags, agricultural residues like bagasse and wheat straw or from waste paper after drying on a battery of steam heated dryers. Paper is a versatile material and finds many uses in our daily life. Traditionally the most common use of paper is for writing and printing, but it also finds various applications as a packaging material, and for large number of industrial and construction processes.

Historically the first use of paper is recorded in ancient Egypt civilisation as a thin paper-like material made from the pith of the papyrus plant, a wetland sedge, grown in the Nile Delta of Egypt. The name paper has therefore originated from the word papyrus. However, the development of pulp and paper making process dates back to 105 AD in China. Over the years, process and technology for paper manufacture has progressed steadily and today in the modern world, paper has linked with cultural and industrial developments. The per capita consumption of paper is often considered as a yardstick of development. The per capita consumption of India stands at only 10.8 kg as against 42 kg in China, 22 kg in Indonesia, 25 kg in Malaysia and 312 kg in the US. This is expected to change in future due to increasing literacy and gradual westernised life style adopted by the Indian populace.

The global paper production recorded in 2013 was 403 million tonnes (399 million tonnes in 2012) and the distribution of its production among the different regions is shown in Figure 16.1. Asia, accounts for 45 per cent (182 million tonnes) of paper production and is by far the largest paper producer. Europe (106 million tonnes) and North America (85 million tonnes) are also significant producers. The annual production of paper by leading paper producing countries is shown in Figure 16.2. China is the largest producer of paper with 105 million tons, followed by USA at 74 million tons. India with 13.5 million tons production is ranked 8th largest producer.

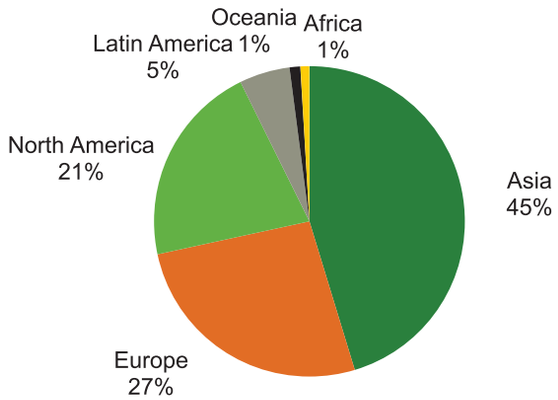

Figure 16.1 Global Paper Production
Source: Pulp and Paper International

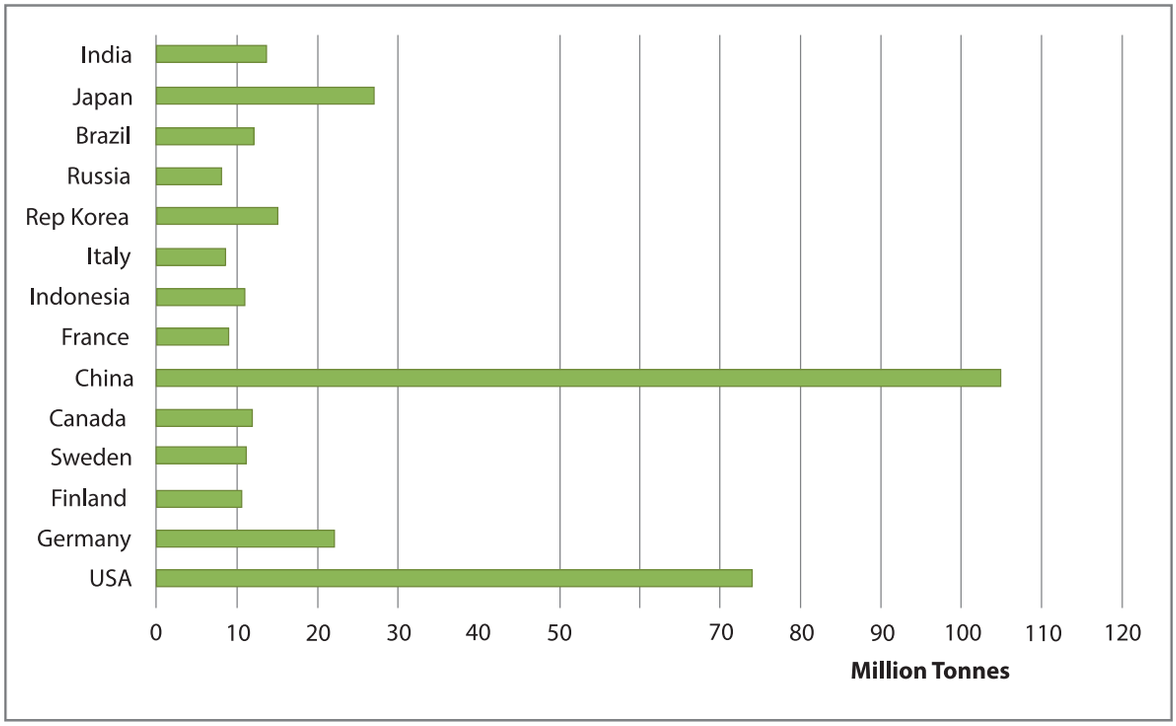

Figure 16.2 Annual Paper Production by leading countries

Source: Pulp and Paper International

The history of mechanised production of paper in India is more than 200 years old, with the first reported setting up of paper mill at Sreerampur, West Bengal in 1812. Presently there are 759 paper industries in India which are divided into three major segments viz., wood based, agro based and recycled fibre / waste paper based, depending on their raw material use. The share of wood based, agro residue and recycled fibre based mills in total production is 31%, 22 % and 47 % respectively. The paper mills are distributed throughout India and the production of different varieties of papers from Indian mills, such as writing/printing, packaging & newsprint produced from different segments of Indian paper industry is about 13.5 million tons.

16.2 Pulp and Paper Manufacturing Processes

The pulp and paper manufacturing can be divided into two major operations.

- **Pulp mill-** It includes raw material preparation (chippers, chip screen, depither etc.), digesters, Brown stock washers, screening and centricleaning, bleach plant and chemical recovery systems for wood and agro based pulp mills. For waste paper based mills the pulp fiber line includes hydrapulper, pulp cleaning systems, deinking and bleaching etc.

- **Stock preparation, paper machine and finishing house-** including refining, addition of fillers and additives, paper machines for production of paper / paperboard & newsprint and finishing house to produce paper rolls and sheets ready for distribution.

The auxiliary operations linked with pulp and paper making include effluent treatment plant, water treatment, power house (boilers and turbines) and other chemical plants like chlorine dioxide and oxygen generation plants etc.

In India, paper mills use diverse raw materials such as wood, agro residues and waste paper. The hardwoods used by the mills are eucalyptus popular, casurina, subabul etc., while the agro residues are wheat straw, bagasse and other annual plants or grasses which are suitable for pulp and paper making. The cellulose fibres in these raw materials are released by chemical pulping actions under strong alkaline conditions at high temperature and pressure, by selectively destroying the chemical bonds between the glue-like substance (lignin) that binds the fibres together. After pulping, the pulp is washed to remove spent chemicals and dissolved lignin in the brown stock washers. The spent pulping liquor separated as filtrate from brown stock washers is concentrated in evaporators and incinerated in specially designed recovery boiler to recover the inorganic chemicals and utilise the heat generated from combustion of organic components as steam for power generation. The washed pulp separated in the brown stock washers is processed further for paper making. The impurities in the pulp such as the foreign materials (sand, dirt etc.) and uncooked/partially cooked fibre bundles are removed from the pulp using screening and centrifugal cleaning equipment. The cleaned pulp is bleached to improve its brightness. Almost 95-97% lignin is removed during pulping and further delignification is continued in bleach plant under milder conditions to reduce cellulose fibre degradation by using milder bleaching chemicals. The bleached pulp produced from wood and agro residues is termed as virgin pulp and is stored in storage towers for paper making. Waste paper is also processed to produce the recycled fibre pulp by using hydrapulpers to disperse the fibers in slurry. The pulp is processed to remove the impurities from the recycled pulp by using different types of centrifugal cleaners and screens. In order to improve brightness of the recycled fibre pulp, it is subjected to deinking and bleaching operations.

In the stock preparation stage, the fibres are subjected to mechanical treatment in refiners. The fibres are refined to make them more conformable by opening up the fibrils from the surface. Fibrillation is an important process operation which makes the fibres suitable for paper making and to obtain the desired paper properties by increasing surface area and entanglement between fibres. At the paper making stage, dyes, strength building resins, or texture adding filler materials may be mixed in the pulp, depending on its intended end product use. For paper making the pulp mixture is dewatered, leaving the fibrous constituents and pulp additives on a wire or wire-mesh conveyor. Additional additives are applied depending on requirement after the sheet-making step. The fibres bond together as they are consolidated through a series of presses and dryers. The final paper product is usually spooled on large rolls for storage. In the finishing house paper is rolled and cut into desired roll sizes or sheets as per customer requirement. The layout of a typical wood based mill is shown in Figure 16.3 and details of major processes for production of pulp and paper are discussed briefly below.

Pulping Process

The pulping of the raw material is performed to liberate the cellulose fibres from the raw material matrix by mechanical, chemical and semi chemical pulping methods. Chemical pulping processes are predominantly used by Indian mills using wood and agro based raw materials.

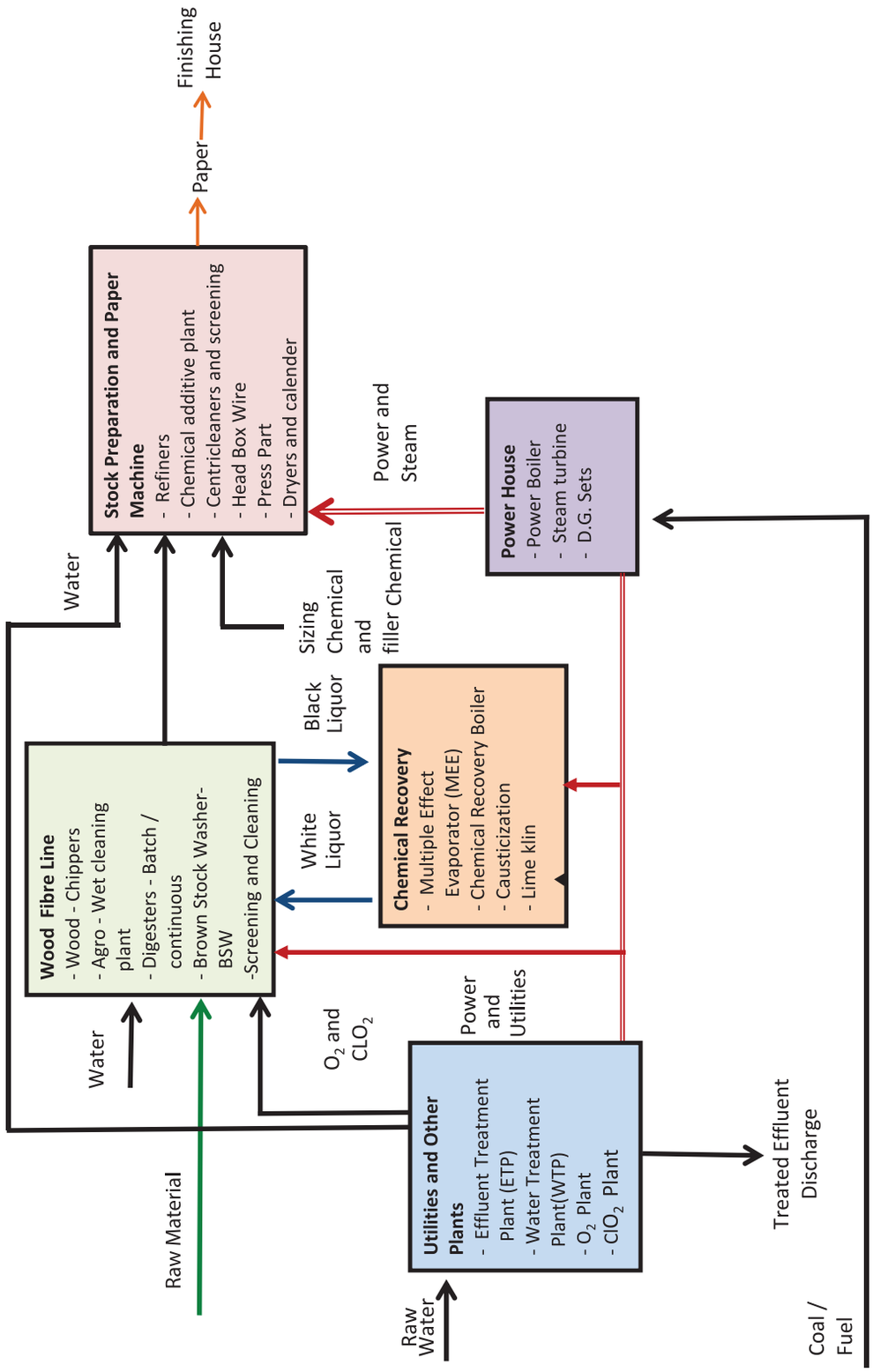

Figure 16.3 Layout of Typical Wood Based Integrated Pulp and Paper Mill

Two types of digesters used for pulping are, batch and continuous digesters. The chips are fed into digesters along with chemicals (NaOH and Na_2S) and digested at high pressure and temperature (7–8 kg/cm^2 and 160-165^{0}C). During pulping, fibres are separated and lignin and other organic/inorganic materials are dissolved. Figure16.4(a&b) illustrate the schematic diagram of batch and continuous digesters.

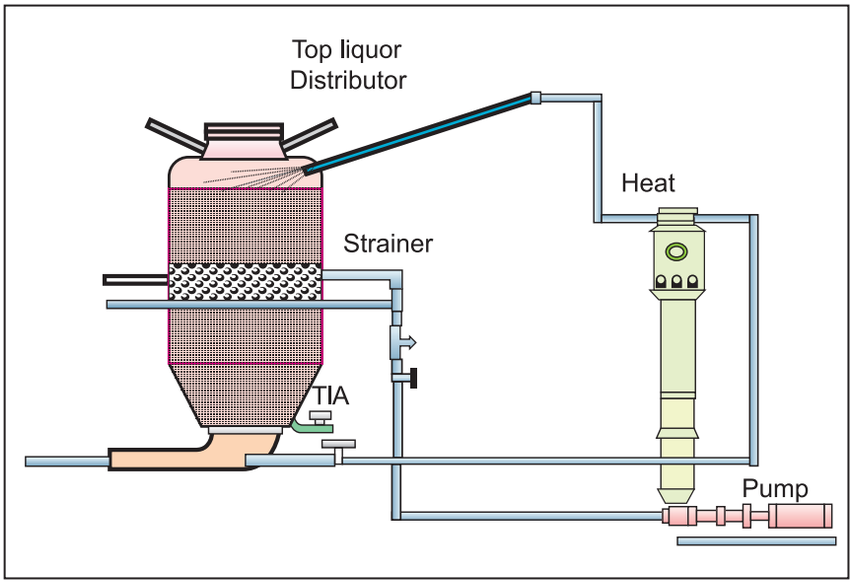

Figure 16.4 (a) Schematic diagram of a Batch Digester

The pulp yield and kappa number are two important variables which are controlled and checked at the end of the pulping process. The kappa number achieved by conventional batch and continuous cooking ranges from 20-30 depending on the type of wood, pulping conditions (temperature, chemical charge and time of cooking etc.) and raw material to liquor ratio etc. maintained during pulping.

Kappa number is the measure of the lignin content (or) bleachability of paper pulp. The International Standard ISO 302:2004 specifies a test method to determine the kappa number. During a test, a known amount of potassium permanganate is added to the sample dispersed in water. Any oxidisable material within the sample, such as lignin will consume permanganate, thus giving a higher kappa number.

Washing, screening and centrifugal cleaning

Subsequent to pulping, the pulp is washed and pulp fibres are separated from the residual pulping liquor in a carefully controlled process known as Brown Stock Washing (BSW). The most common washing method employs a series of counter current vacuum drum washers to displace liquor with minimum dilution. A number of efficient alternative options such as rotary pressure washer, horizontal belt washer, twin roll press etc. are available now a day and are being used by modern mills for efficient washing of pulp.

The advanced rotary pressure washers and horizontal belt washers offer a compact design and in a single washer 2 to 3 displacement stages can be operated. The solid of the extracted liquor is also higher than the conventional vacuum washers. The twin roll wash presses or extraction presses are

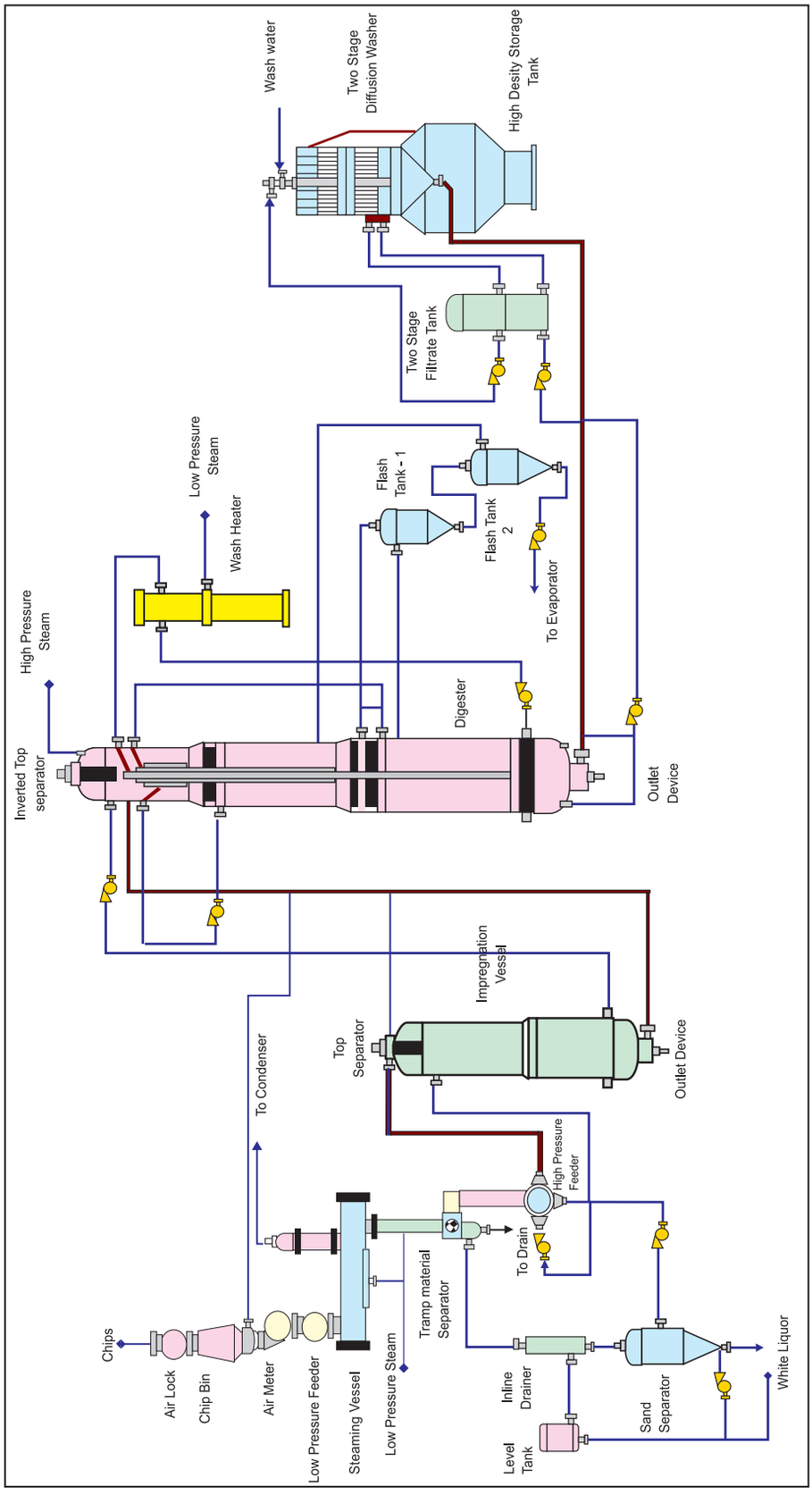

Figure 16.4 (b) Schematic diagram of a Continuous Digester

capable of achieving discharge consistencies around 30-40%. They are quite competitive compared to conventional vacuum washer systems and suitable particularly for pulps that are difficult to permeate, i.e., where a displacement-type washer would not be an appropriate choice.

After washing, the pulp is screened and cleaned. Screening of the pulp is performed to separate coarse rejects like knots and uncooked chips from the pulp, remove stones and junk in order to avoid wear and damage of machines. Screening also acts as a "guard" for eventual hard cooks as it is used to remove the shives from the pulp in order to achieve a pulp free from bigger shives. The major types of stock screens employed are vibratory, gravity centrifugal, and pressure (centrifugal or centripetal).

The centrifugal cleaners are also used for pulp cleaning. Centrifugal cleaners remove unwanted particles from pulp and paper stock by a combination of centrifugal force and fluid shear on the basis of density differences and particle shape. All centrifugal cleaners work on the principle of a free vortex generated by a pressure drop to develop centrifugal action. The pressure for cleaning is generated from the feed pump and the stock enters the cleaner tangentially imparting a rotating motion through the inlet scroll guides. Due to density difference the long fibre useful for paper making are carried inward and upward to the accept stock outlet. The dirt, held in the downward current, is moved toward the tip and removed from the pulp stock. Figure 16.5 shows the flow sheet of a bleached kraft pulp street.

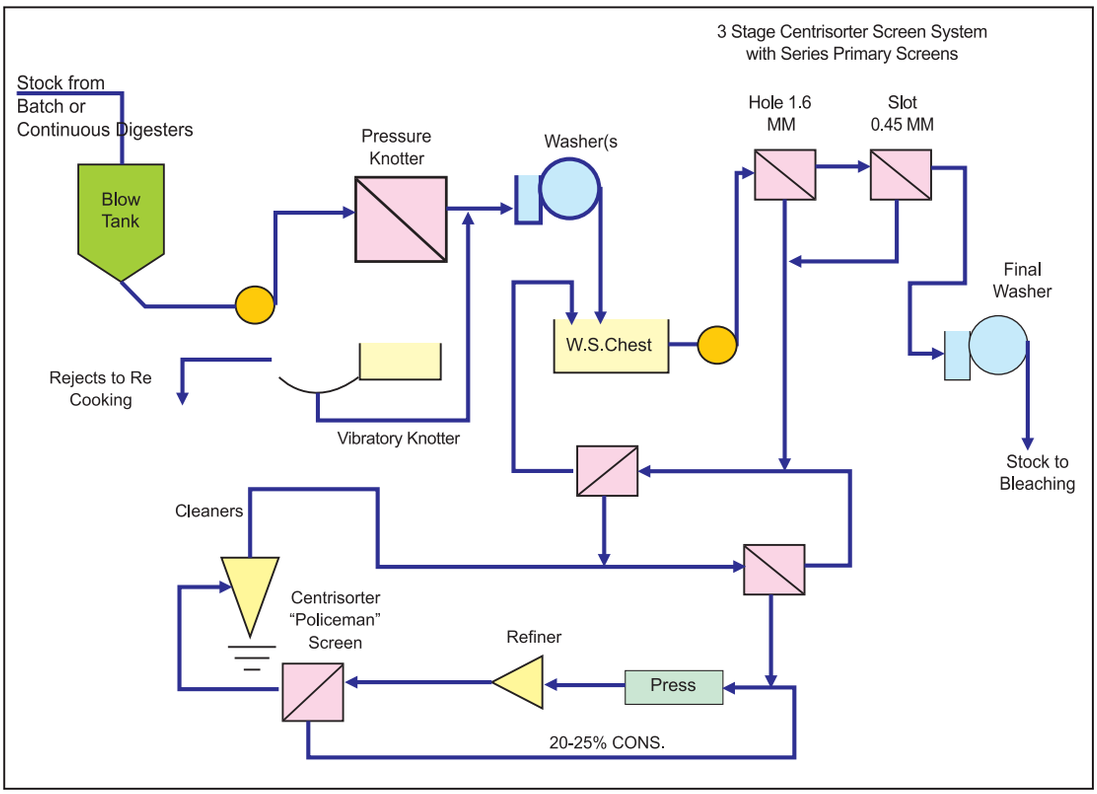

Figure 16.5 Basic flow sheet of low – yield bleachable kraft pulp

Extended Delignification

To control bleach chemical cost, it has always made good sense in the production of bleachable grade pulps to de-lignify the raw material as much as possible during cooking, and thereby minimize the amount of residual delignification required during bleaching stage. More recently, with attention focused on reducing the discharge of chlorinated organic compounds from bleach plants, the need to de-lignify pulp more completely prior to conventional bleaching has become essential. Today with the use of extended delignification, it is possible to obtain pulps with kappa number as low as 12-15 compared to the conventional pulps with kappa number 20-30. Low kappa number of the pulp signifies higher de-lignification in the pulp.

Modern batch digesters are upgraded with **Rapid Displacement Heat (RDH)** batch cooking or Super batch cooking systems to obtain pulp of high quality with improved energy efficiency. The operations in a modified pulping systems are chip filling, filling of warm impregnation liquor, hot liquor filling for heating and cooking by displacement of warm impregnation liquor, heating and cooking by recirculation of hot liquor using small amount of steam in heat exchanger and finally displacement of the hot spent liquor with cool liquor by pumping into the bottom of the digester on achieving the desired delignification. Displacement of hot liquor from digester reduces the temperature of the pulp below 100^0 C.

Oxygen delignification (ODL) is also one of the methods used for reducing the lignin content of pulp before conventional bleaching. In modern pulp mills, the digesters equipped with extended de-lignification (i.e., RDH/super batch pulping) are followed by ODL to obtain low kappa pulps and in these mills the bleach chemical demand is significantly reduced. Although technically independent, the oxygen delignification stage is compatible with the kraft recovery process because its caustic effluent can be added to black liquor and processed through the recovery furnace. In India most of the large wood based mills have oxygen delignification stage prior to bleaching stage.

Bleaching

In order to obtain the white bleached pulp, unbleached pulp is treated with bleaching chemicals like chlorine, chlorine dioxide, hydrogen peroxide, oxygen and ozone etc. Modern bleaching involves continuous sequence of process stages utilizing different chemicals and conditions in each stage, usually with washing provided in between stages. Since elemental chlorine is known to be a major contributor of chlorinated organic compounds (also known as “adsorbable organic halides” or AOX) to the environment, therefore as a Corporate Social Responsibility there has been gradual decline in use of elemental chlorine and increase in substitution / partial substitution of chlorine by chlorine dioxide, known as Elemental Chlorine Free (ECF) bleaching.

Bleaching is achieved through a continuous sequence of process stages utilizing different chemicals and conditions in each stage, usually with washing between stages. The conventional symbols for various bleaching stages are given in Table 16.1.

Table 16.1 Symbols for various bleaching sequences

C Chlorination	PO Oxygen Peroxide
H Hypochlorite	Pa Peracetic Acid
E Alkaline Extraction	Z Ozone
D0, D1, & D2 Chlorine Dioxide (Initial Delignification, First and Second Brightening)	Q Chelation
O Oxygen Delignification	N Neutralization
E_O Oxygen Enriched Extraction	X Enzymes
P Peroxide	Y Hydrosulfite

The common sequences employed in Indian pulp and paper mills are CEPHH, CEDED, (C+D)(E_o) DED (in this sequence dioxide is added sequentially to the chlorination stage and oxygen utilized in the extraction stage), D(Eo)DED or D(EoP)DED.

Chemical Recovery

The filtrate containing spent pulping chemicals, generated from pulp washers is processed in the recovery section to recycle cooking chemicals for pulping and to utilize energy from the incineration of organic residuals. This leads to minimization of air and water pollution from pulp mill. The steps involved in chemical recovery operation, starting with generation of “weak black liquor” from the brown stock washers, are as follows.

- Concentration of the spent weak black liquor in multiple effect evaporators to form "strong black liquor" (45-50% solids) and further concentration to form "heavy black liquor" (60-70% solids).
- Salt cake addition to make up soda loss.
- Incineration of liquor in the recovery furnace.
- Dissolution of smelt from the furnace to form green liquor.
- Causticiziation of green liquor with lime to form white liquor.
- Burning of lime mud in lime kiln to recover lime.

A typical operating sequence of chemical recovery is illustrated in Figure 16.6. The concentration of black liquor is carried out in the Multiple-Effect Evaporators (MEE), which is a series of live steam or vapour heated bodies operated at different pressures to utilize the vapor from one evaporator body as the steam supply to the next body. Different designs of evaporators are used, but in all cases a main objective is to minimize fouling and scaling and achieve higher efficiency. In Indian mills the most common types of evaporator used are the long tube vertical (LTV) and the free flow falling-film (FFFF) design evaporator bodies. FFFF evaporators are more efficient.

Incineration of the heavy black liquor is performed in the recovery boiler furnace. The liquor droplets dry and partially pyrolysis before falling onto the char bed at the bottom of the furnace, where incomplete combustion in air deficient environment causes carbon and carbon monoxide to act as

reducing agents and converts sulfate and thiosulfate to sulfide. The heat at the char bed is sufficient to melt the sodium salts, which then flow by gravity through water-cooled spouts to the smelt dissolving tanks where the smelt is converted into green liquor by addition of weak wash.

In the causticization operation the green liquor is reacted with lime (CaO) to form NaOH or white liquor. The white liquor is clarified to remove precipitated "lime mud" ($CaCO_3$) for cooking. Auxiliary operations include washing of both the dregs and lime mud for soda recovery, and the calcining ("reburning") of lime mud to regenerate lime. The lime mud is converted to lime in the the lime kiln by calcining (or lime "reburning") for its reuse in the causticizing process.

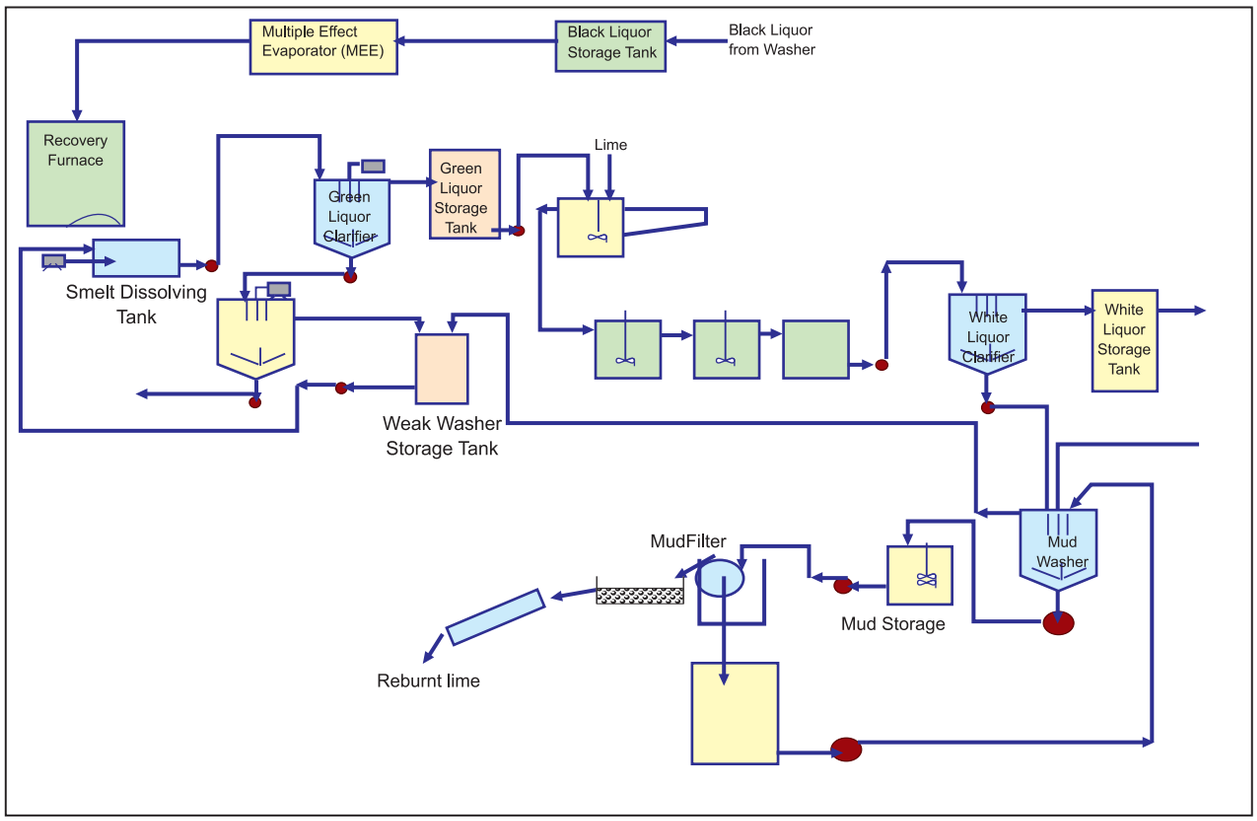

Figure 16.6 A typical Chemical Recovery Operation

Stock Preparation

Refining is the most important unit operation in stock preparation in which the fibers are subjected to mechanical action to develop their optimal paper making properties with respect to the product being made. During refining the amount of energy absorbed by the pulp is the major factor affecting the change in pulp properties; but the manner in which the work is carried out is also a significant determinant. Refining carried out at low intensity produces greater fibrillation, less cutting, and more satisfactory development of fiber properties. In other words, a gradual, step by step application of mechanical energy to the fibers provides the optimal treatment, in contrast to that produced by a more abrupt and concentrated application of the same amount of energy. Two major types of refiners used for stock preparation are disc refiners and conical refiners. Disc refiners are a more recent development,

and are available in a wide variety of designs and disc patterns. Efficient refining requires control of refining conditions, selection of right equipment and its configuration etc., which results in energy efficiency during refining.

A wide variety of mineral and chemical agents are added to the stock, either to impart specific properties to the paper or to facilitate the paper making process. The fibrous and non-fibrous furnish components are blended to form the paper making stock. Various wet end additives are added to achieve desirable brightness, opacity, wet strength, colour and shade.

Paper Machine

In paper machine the refined pulp stock is converted into a wet web from the pulp suspension which is further dried to get the paper or paper board. Today high speed machines are available to produce upto 8 to 10 meter wide web of paper at speed above 1200 m/min. Mills use variety of machines running on Mono Glazed (MG) cylinder, Fourdrinier, twin wire and duo-former technologies. In terms of energy utilization in paper machine, removal of moisture from the paper web by vacuum dewatering and press drying in steam heated dryers are the major areas which require extensive energy input. For higher energy efficiency and energy cost reduction it is desired that maximum possible moisture removal should be achieved in these steps by following best practices and modern technologies. A basic fourdrinier type machine, common in Indian mills is shown in Figure 16.7.

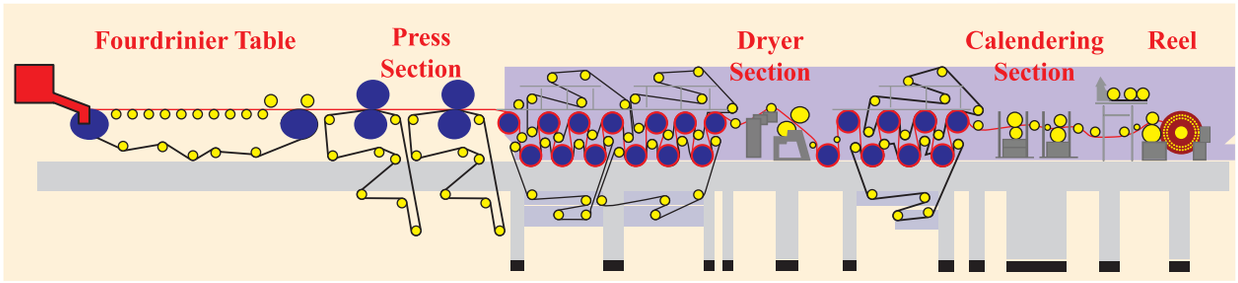

Figure 16.7 The Basic Fourdrinier Type Machine

In the fourdrinier table or wire part, the paper web is formed by removal of water and the paper web with dryness upto 22% can be obtained by proper operation of various drainage elements in the wire part.

The wet web is pressed to remove moisture from the paper sheet by mechanical action between the press nips. Pressing operation is considered an extension of the water-removal process that starts on the wire. It is far more economical to remove water by mechanical means than by evaporation, so the papermaker always looks for means to improve pressing efficiency and reduce the evaporative load in the subsequent dryer section. Modern machines are equipped with shoe press which facilitates extended nip dwell time due to its special design. This ensures high performance even at a high machine speed. The machine rebuilds with shoe press in a large number of machines have reported significant energy saving due to high moisture removal leading to steam saving in the dryers.

After pressing, the sheet is passed through the dryer section where the residual water is removed by evaporation on a series of large diameter, rotating, steam-filled cylinders. The dryer section is most costly to operate because of high energy consumption in drying.

The steam economy of the paper dryers is mostly affected due to the steam/condensate and air handling systems. The dryer fabrics also play vital role in energy efficiency. In modern paper machines synthetic fabrics with permeable construction are used to provide ventilation by freely carrying air into and out of the pockets between dryers.

Air has important role in paper drying process, therefore the dryer hood arrangement is also important for water evaporation. In general 3.5 to 10 kg air is utilized for each kg of water evaporated in the dryer section depending on the type of hood arrangement. All modern machines are equipped with hoods having heat recovery system to recover heat from the hot, humid exhaust air into the fresh, ambient supply air.

Proper control and optimization of the paper making operations such as approach flow systems, head box, drainage elements on the wire, forming fabric, type and layout of presses, dryer condensate removal, pocket ventilation and hoods etc, contribute significantly towards product quality and energy savings.

16.3 Energy Consumption Pattern

Energy is an essential input and a major cost driver in paper manufacturing. Energy constitutes 20-30% of the cost of manufacturing and it is the only cost variable which can be controlled to make paper industry cost competitive. Figure 16.8 illustrates the % cost break-up in a wood based mill producing writing printing paper.

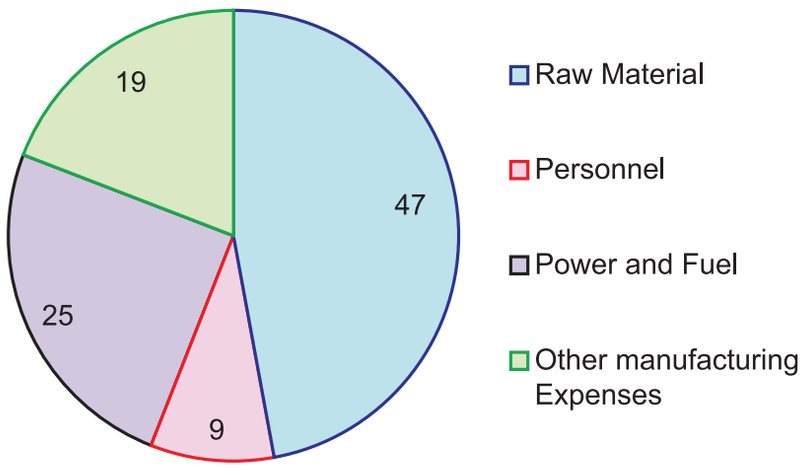

Figure 16.8 % Cost break-up in Indian mill producing writing printing paper from wood

Various forms of primary and secondary energy sources are used in pulp and paper making operations. Among the various energy sources, such as fossil fuels, agro residues and biomass generated within

the industry, coal is the largest used fuel. Other fuels such as bagasse pith, rice husk, coconut shell, groundnut shell etc. are also used depending on their availability by the mill. Some internally generated combustible wastes, such as black liquor, pith, chipper dust, sludge etc are also used as fuel. These are generated from raw material preparation, pulping and paper making operations. Black liquor, the spent pulping liquor, accounts for more than 80% of the biomass-based fuel used in the paper industry. Steam and power are the other major secondary forms of energy used in pulping and paper manufacturing processes.

Details of the energy consumption in wood, agro and RCF based mills are presented below.

Energy Consumption in Integrated Wood and Agro Based Mills Using Conventional and Non-conventional Chemical Recovery Systems

Unit operations in Integrated wood and agro based mill are generally similar, but technological status of agro based integrated mill varies based on the size of the mill. In large agro based mills (above 300 TPD) the processes used are based on State-of-Art technologies, however, in smaller sized integrated agro based mills, with capacity up to 150 TPD, technological level is relatively obsolete. The section wise breakup of specific steam and power required for unit production of paper from a modern integrated wood /agro based mills using conventional recovery system integrated with steam turbines in power house is presented in Table16.2.

The agro based mills below the capacity of 100 TPD utilize non-conventional chemical recovery systems on account of its economic viability. Table 16.2 also presents the section wise breakup of steam and power in an integrated agro based paper mill using Non-conventional chemical recovery system with power house.

Table 16.2 Section-wise Energy Consumption in an Integrated Wood/Agro Based Mill Using Conventional Recovery System and the Integrated Agro Based Mill using Non-Conventional Recovery System. (Source: CPPRI)

S. No.	Section	Wood/Agro Based Mill Using Conventional Recovery System		Agro Based Mill Using Non-Conventional Recovery System	
		Power Consumption, kWh/ton of paper	Steam Consumption, ton/ton of paper	Power Consumption, kWh/ton of paper	Steam Consumption, ton/ton of paper
1	Pulp Mill (including Raw material cleaning, Digester, screening & washing and bleach plant)	300-325	1.2-2.5	300- 325	1.2-2.5
2	Recovery Section	250-300	3.0-3.5	75 -100	1.0-1.7
3	Stock Preparation & Paper Machine	350-450	2.5-4.0	350 – 450	2.5-3.5

4	Effluent Treatment Plant	75-100	-	75 – 90	-
5	Power Generation Plant	150-200	6.0-6.5	125 -200	6.5-7.0
6	Others (including ClO_2, O_2 plant, office, street light)	100-125	-	100 -125	-
	Total	**1225 - 1500**	**12.7- 16.5**	**1025 - 1290**	**11.2 - 14.7**

Energy Consumption in Recycled Fiber (RCF)/Waste Paper Based Mills

The RCF based mills can be grouped into two categories depending on the manufacturing process and the final product. These are (1). Mills producing white varieties using De-inking and bleaching systems and (2). Mills without De-inking systems.

The overall energy consumption in the major unit operations of RCF based mill producing white varieties using De-inking systems as well as the mill producing white and unbleached varieties without De-inking system is presented in Table 16.3.

Table 16.3 Section-wise Energy Consumption of RCF Based Mill Producing White Grade of Paper Using Deinking and Oxidative Bleaching Systems as well as the Mill Producing White and Unbleached Varieties Without De-inking System. (Source: CPPRI)

S. No.	Section	RCF Mill using Deinking and Bleaching Systems		RCF Mill without Deinking and Bleaching Systems	
		Power Consumption, kWh/t paper	Steam Consumption, t/t paper	Power Consumption, kWh/t paper	Steam Consumption, t/t paper
1	Pulper and pulp cleaning (HD Cleaner & Centrcleaner, De-Inking & oxidative bleaching)	300-350	0.50-0.70	100- 130	0.25
2	Stock Preparation & Paper Machine	350-450	2.10 -3.00	225- 250	2.1-3.0
3	ETP	40-45	-	40-45	-
4	Power Generation Plant	100-110	6.00-6.60	-	-
5	Others (WTP, Office, Street lights, Workshops)	40-45	-	25- 40	-
	Total	**830-1,000**	**8.60-10.30**	**390-465**	**2.35-3.25**

16.4 Material and Energy Balance Calculations in Pulping and Paper Making

Pulping

(i) Chemistry of Pulping

The two commonly used processes by Indian pulp & paper mills for pulping are soda and kraft processes. The soda process is used by medium sized agro based mills based on non-conventional recovery and the kraft process by large wood and agro based mills using conventional chemical recovery systems. In soda process sodium hydroxide is the major chemical used for cooking whereas in kraft process, cooking liquor or white liquor, an aqueous solution of sodium hydroxide (NaOH) and sodium sulphide (Na_2S) is used for pulping. The hydroxyl ion (OH^-) and the hydrosulfide ion (SH^-) are the active components in the cooking liquor. They originate from NaOH and Na_2S as follows.

$$NaOH \rightarrow Na^+ + OH^-$$

$$Na_2S \rightarrow 2\,Na^+ + S^{--}$$

$$S^{--} + H_2O \rightarrow SH^- + OH^-$$

The concentration and total charge of the SH^- and OH^- ions are the key elements in all kraft pulping reactions.

(ii) Terms Used in Pulping

Liquor to raw material ratio

Liquor to raw material ratio is a term that describes the amount of total liquor per unit weight of dry raw material in the digester. The moisture content of the raw material is also included in total liquor. The liquor to raw material ratio varies for each type of raw material and typically varies between three to five.

Alkali Charge

Two commonly used terms to indicate the chemicals charged in digester are total and active alkali. Total alkali indicates concentration of sodium salts in white liquor on basis of equivalent amount of sodium oxide (Na_2O) or as sodium hydroxide (NaOH) and it includes NaOH, Na_2S, Na_2CO_3, ½ Na_2SO_3 present in white liquor. The active alkali includes only NaOH and Na_2S, which are the active chemicals for pulping. Both total and active alkali are expressed as percentage of dry lingo-cellulosic raw material weight are then referred to as total or active chemical / alkali charge.

Cooking Cycle

Various stages of cooking cycle are pre-steaming, raw material impregnation, cooking and blowing. Total cooking cycle is 3-4 hrs for a batch digester. The cooking however, takes place at a constant temperature of approximately 160 to 165^0 C for one to two hours. Most of the delignification takes place in the cooking stage.

Kappa Number

Kappa Number is a measure of residual lignin in the pulp. It is helpful to determine the degree of delignification achieved during cooking and to find out the bleach chemical demand during bleaching.

Example 16.1: Pulp cooking and steam requirements for digestion

In a digester with 42.5 MT digester shell, 100 MT of chips with 50% moisture are cooked at 170^0C by maintaining 1:4 bath ratio. The digester is filled with 112.5 MT of white liquor at 90^0C and 37.5MT of black liquor at 85°C. Find out the total steam requirement per cook and steam per tonne of the chips and pulp.

- Sp. heat of chips at 50% moisture is 0.33, kcal/kg°C at 35°C
- Sp. heat of white liquor =0.91 kcal/kg °C at 90°C
- Sp. heat of Black liquor =0.94 kcal/kg° C at 85°C
- Sp. heat of digester shell =0.117 kcal/kg°C at 90°C.
- Temperature after blow = 105°C
- Radiation losses from digester = 8%
- Heat of reaction = -10% (exothermic)
- Sp heat of cooking liquor after cooking = 0.92 kcal/kg°C
- Pulp Yield= 40%
- Allowance for vent steam and heat blow = 5%

Solution:

The heat requirement for each component in digester is calculated as

S. No.	Components	Mass (M) (MT)	Sp. Heat(Cp), (kcal /kg° C)	ΔT (^{0}C)	Heat Requirement, Q= M Cp Δ T (x10^3 kcal)
1	Dry Chips	50	0.33	170-35=135	50 x .33 x135 = 2227.5
2	Chip Moisture	50	1	170-35=135	50x1x135=6750
3	White Liquor	112.5	0.91	170-90=80	112.5 x 0.91 x80= 8190
4	Black liquor	37.5	0.94	170-85=85	37.5 x .94 x 85= 2996.25
5	Dig shell	42.5	0.11	170-90= 80	42.5 x 0.11 x 80 =3978
	Total Heat Requirement ∑Q = 24141.75 x10^3 kcal				

Radiation losses @ 8 % of ∑Q = 24141.75 x .08 = + 1931.34
Heat of reaction @ -10% of ∑Q = 24141.75 x 0.1 = - 2414.175
Total Heat (∑Q) after correction of radiation losses and heat of reaction = 23658.92x10^3 kcal

Add allowance for vent steam and heat blow =5%
So Actual heat (∑Q) after addition of allowance for vent steam and heat blow

$$= 23658.92 \times 10^3 \times 1.05$$
$$= 24841.86 \times 10^3 \text{ kcal}$$

$$Steam / Cook = \frac{24841.87 \times 10^3}{490.38 \text{ (Latent heeat of stream at } 170^oC)} = 50.7 \text{ MT (X)}$$

$$Steam / MT \text{ of OD Chip} = \frac{X}{\text{Qty. of raw material per cook}}$$

$$= \frac{50.7}{50} = 1.02 \text{ MT/tonne of Chip}$$

$$Steam / T \text{ of OD Pulp} = \frac{X}{\text{Pulp produced per cook}}$$

$$= \frac{50.7}{50x0.4} = 2.535 \text{ MT/tonne of Pulp}$$

Paper Machine

The paper making process is essentially a very large dewatering operation where a diluted solution of pulp suspension with less than 0.5 % fibre solids is formed into a web. Regardless of the nature of the pulp, whether it is chemical, mechanical or recycled from softwood, hardwood, agricultural residue, the basics of paper making are similar.

Complete process of water removal on the paper machine is carried out by four principal operations: 1) By “free drainage” on the fourdrinier, 2) By “suction” on the wire part, 3) By “pressing” on the press part and 4) by “evaporation” on the dryer part. The majority of the water is removed in the wire part to form the wet web of the paper. The sheet dryness after couch is 20-22%. The wet paper sheet after wire part is pressed and consolidated into a dense sheet with removal of moisture by pressing. After mechanical limits of water removal in the press is reached, the wet sheet enters the dryer part, where the dryness content of the sheet is increased by evaporation. The dryer section of a paper machine removes between 1.1 to 1.3 kg of water per kg of paper compared to 200 kg and 2.6 kg in the forming and press section respectively. It is significantly more expensive to remove water in the dryers than the forming and press section. The relative cost of dewatering are, forming section 10%, press Section 12% and dryer section 78 %. The dryer section is by far the largest consumer of thermal energy of the paper machine.

In this section some examples of the paper de-watering and steam requirements for paper drying are discussed.

Example 16.2:

A multicylinder fourdrinier paper machine is producing 100.0 tonnes cream woven paper with 6% moisture content without break per day. The consistency of the fibre suspension inside the head box is 0.5% and the dry content leaving the couch and the last press is 20% and 35% respectively. Determine percentage of water removed,

a. By free drainage and suction on the wire part
b. By pressing on the press part
c. By evaporation from the dryers
d. Total quantity of water evaporated from the dryers per day

Neglect fibre losses, if any, with the back water.

Solution:

Basis: 100 tonnes /day production of cream woven paper
Hence, quantity of dry content carried by the paper per day = 0.94 x 100 = 94 tonnes

S.No.		Leaving the Slice	Leaving the couch	Leaving the last press	Leaving the last dryer
1	Dry Content	0.5 %	20%	35%	94%
2	Fibre –water ratio	0.5:99.5	20:80	35:65	96:6
3	Tonnes of water carried over per day	$\frac{99.5}{0.5}$ x94 = 18706	$\frac{80}{20}$ x94 = 376	$\frac{65}{35}$ x94 = 174.57	$\frac{6}{96}$ x94 = 6.00

4	Tonnes of water removed per day	Wire Part 18706 – 376 =18330	Press Part 376 – 174.57 = 201.43	Dryer Section 174.57 - 6 =168.57
5	% of water removed	$\frac{18330}{18706}$ x 100 = 97.98%	$\frac{201.43}{18706}$ x100 =1.07%	$\frac{168.57}{18706}$ x100 =0.90 %

Ans:

a) Water removed on the wire – part = 97.98%

b) Water removed by the presses = 1.07%
c) Water evaporated by the dryers = 0.90%
d) Total quantity of water evaporated by the dryers per day = 168.57 tonnes

Example 16.3:

The average rpm of the dryer of a paper machine, producing 70 GSM Maplitho paper of 70% moisture, is 65. The deckle of the paper on the reel is 3.3 meters. The diameter of the drier is 1.5 meters. The steam pressure inside the drying cylinder is kept constant at 2.5 kg/cm^2. If the sheet dryness after presses is reduced from 40% to 35%, Calculate:

a. Amount of additional water evaporated per day for paper at 93% dryness.
b. Amount of additional steam required per day.

Given that latent heat of evaporation of water at 100^{o}C from the paper at zero pressure is 540 kcal/kg and that of the steam in the drier at 2.5 kg/cm^2 at 126 ^{o}C is 522 kcal/kg.

Solution:

Given, diameter of the dryers = 1.5 meters

Machine Speed = $\prod$ x diameter (m) x rpm = $\frac{22}{7} \times 1.5 \times 65$ = 306.4 meters / min

Production = speed (m/min) x GSM x deckle (m) x minutes per day/ (1000 x 1000)
= (306.4) (70) (3.3) (23 x 60) / 1000 x 1000
= 97.67 Tonnes / day (Considering production period 23 Hrs a day)

a) **Evaporation when dryness after presses is 40 %**

Moisture in wet-web entering the driers
(kg of water / kg of dry fibre) = 60/40 = 1.5

Moisture in finished paper,
(kg of water / kg of dry fibre) = 7 / 93 = 0.075

Evaporation per kg of dry fibre = 1.5 – 0.075 = 1.425

Evaporation per kg of paper produced at 93 % dryness = 1.425 x 0.93
= 1.325

Evaporation when dryness after presses is reduced to 35%

Moisture in wet –web entering the dryers = 65/35 = 1.857
(kg of water/ kg of dry fibre)

Moisture in finished paper will remain same

Therefore evaporation per kg of dry fibre	= 1.857 – 0.075 = 1.782
Hence Evaporation per kg of paper produced	= 1.782 x 0.93 = 1.657
Additional water evaporated	= 1.657 – 1.325 = 0.332
Thus, additional water evaporated per day (kg of water evaporated / kg of paper produced)	= 97.67 x 0.332 **= 32.46 Tons**

b. Since the steam pressure inside the drying cylinders is kept constant at 2.5 kg/cm^2, therefore additional steam requirement shall be only due to the change in the dryness after presses.

At 40% dryness, evaporation = 1.325 kg / kg of paper produced

At 2.5 kg/cm^2

kg of steam condensed / kg of water evaporated $= \frac{540}{522} = 1.034$

Therefore, kg of steam condensed / kg of paper produced = 1.325 x 1.034 = 1.37

Similarly,

At 35% dryness evaporation = 1.657 kg / kg of paper produced

kg of steam condensed / kg of paper produced	=	1.657 x 1.034 = 1.71
Additional steam requirement	=	1.71 - 1.37 = 0.340
Hence, additional steam required per day (kg of steam / kg of paper produced)	= =	97.67 x 0.340 33.21 Tonnes

Conclusions

The technological complexity and variation in process layouts in different pulp and paper mills based on wood, agro and RCF, leads to wide difference in energy consumption for production of similar products. A large number of energy management, best practices, advanced process control and State-of-Art technologies have been adopted by the mills, however still there is ample scope for improvement and energy efficiency enhancement in pulp and paper manufacturing. The energy auditors with sound understanding of pulp and paper mill processes can contribute significantly towards energy conservation, as the process optimization often leads to large savings.

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Chapter 17

Performance Assessment of Fertiliser Industry

Learning Objectives

In this chapter you will learn about

- Fertiliser manufacturing process
- Distinct process steps for manufacturing urea
- Energy flow in a typical ammonia / urea fertiliser complex
- Material & energy flow of an ammonia / urea fertiliser complex
- Steam balance in an ammonia / urea complex.

Energy Auditors

After successfully completing this chapter you will be able to complete the following tasks:

- ✓ **Understanding the process of fertiliser manufacturing**
- ✓ **Estimate steam balance and energy balance in an ammonia / urea complex**

17. ENERGY PERFORMANCE ASSESSMENT OF FERTILISER INDUSTRY

17.1 Introduction

The inorganic, organic, natural or synthetic chemical elements that provide nutrient for the growth of plant are generally considered as fertiliser. The nutrients are categorized further as Primary, Secondary and Micronutrient based on their requirement by soil. The primary nutrients are: Carbon, Hydrogen, Oxygen, Nitrogen, Phosphorus and Potassium. Of the primary nutrients, Carbon, Hydrogen and Oxygen are supplied through air and water while Nitrogen (N), Phosphorus (P_2O_5) and Potassium (K) are supplied through chemical or mineral fertilisers. There are three secondary and more than 13 micro nutrients that also play an important role in plant growth.

Indian Fertiliser Industry is the third largest producer of fertiliser in the world. There are 152 plants in operation. This is comprised of 30 urea, 19 DAP and NP/NPK complex, 91 single super phosphate, 10 ammonium sulphate, one calcium ammonium nitrate and one ammonium chloride fertiliser plants. India produced 38 milllion tonnes of fertiliser products during 2013-14. Urea accounts for 60% of total production of fertilisers.

Production of urea is highly energy intensive and accounts for almost 85% of the energy requirement of the fertiliser sector. Contribution of other fertilisers is very small compared to urea production. For example, the weighted average energy requirement for production of urea is around 6.2 Million kcal/ tonne of urea compared to 0.25 Million kcal/tonne for a Complex fertiliser. Further, for production of urea, 80% of the energy is consumed in production of ammonia. Therefore, the scope of energy conservation is maximum in ammonia plants. Hence, efforts for conservation of energy are focused on ammonia plants.

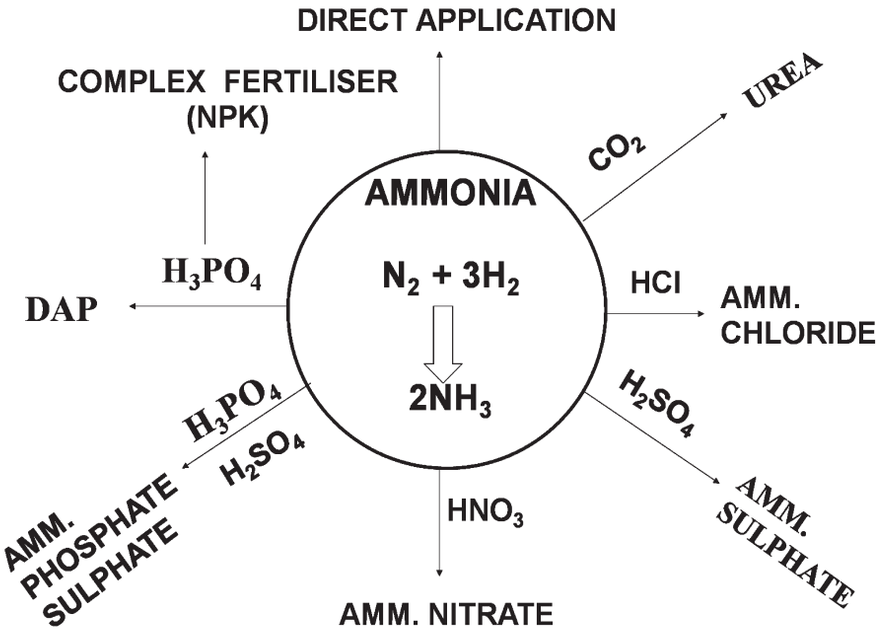
During sixties and mid-seventies, emphasis was on establishing fertiliser production capacity by putting up new plants. These plants were based on raw materials readily available at that time. In 1970's due to addition of petroleum refineries, naphtha became the preferred feed stock for fertiliser plants. Again due to surplus availability of fuel oil, six plants were constructed based on this feedstock. During this period, fertiliser industry being in early stage of development in India, focus was on reliability of plants and focus was not on energy efficiency.

However, with passage of time, energy became scarce and expensive. This necessitated revamp of older plants, by changing over to Natural Gas (NG) as feedstock and upgrading technology. During eighties, large quantity of natural gas was available from Bombay High. This provided more efficient and cheaper feedstock. By this time, the fertiliser technology had made significant advancement. This was the period during which the fertiliser industry grew very rapidly. During nineties, the technological developments in manufacturing of ammonia / urea were at their best. Significant fertiliser production capacity was added with more efficient plants. However, due to unabated increase in price of hydrocarbon products, reduction in energy cost became the ultimate focal point of the industry.

As a result, a number of retrofits were implemented by old plants like change in ammonia converter basket, better heat recovery from reformer furnace exhaust, gas turbines coupled with HRU etc. These measures brought down energy consumption significantly. A number of old vintage plants not operating at reasonable levels due to obsolete technology and equipments became unviable and were shut down in 1990s.

17.2 Fertiliser Manufacturing Process

Ammonia is the building block of all nitrogen containing fertilisers. Total installed ammonia capacity in India is 13.70 tonnes. Almost 95% of ammonia produced is used for making urea and the balance is used for manufacture of complex fertilisers. Urea is the major nitrogenous fertiliser and accounts for 83 % of the total nitrogen production. Other nitrogenous fertilisers are ammonium sulphate (AS) and calcium ammonium nitrate (CAN), which are produced in smaller quantities. Among complex fertilisers, di-ammonium phosphate (DAP) and various grades of NP/NPK are produced. Apart from these, single super phosphate (SSP), also contributes to the phosphatic production.



17.2.1 Ammonia

The basic process of ammonia synthesis is Haber process where ammonia is synthesized by catalytic reaction of nitrogen and hydrogen at high pressure. Nitrogen is derived from atmosphere either by air separation or using air for combustion of hydrocarbons generating a mixture of nitrogen, hydrogen and carbon oxides. Hydrogen can be derived from a variety of raw materials including water (electrolysis), heavy hydrocarbons (partial oxidation), coal/coke (gasification), light hydrocarbons (steam reforming). In all cases of fossil fuels, part of hydrogen produced is derived from water used as steam.

Distinct process steps for manufacturing ammonia using hydrocarbon reforming process are described below (Figure 17.1).

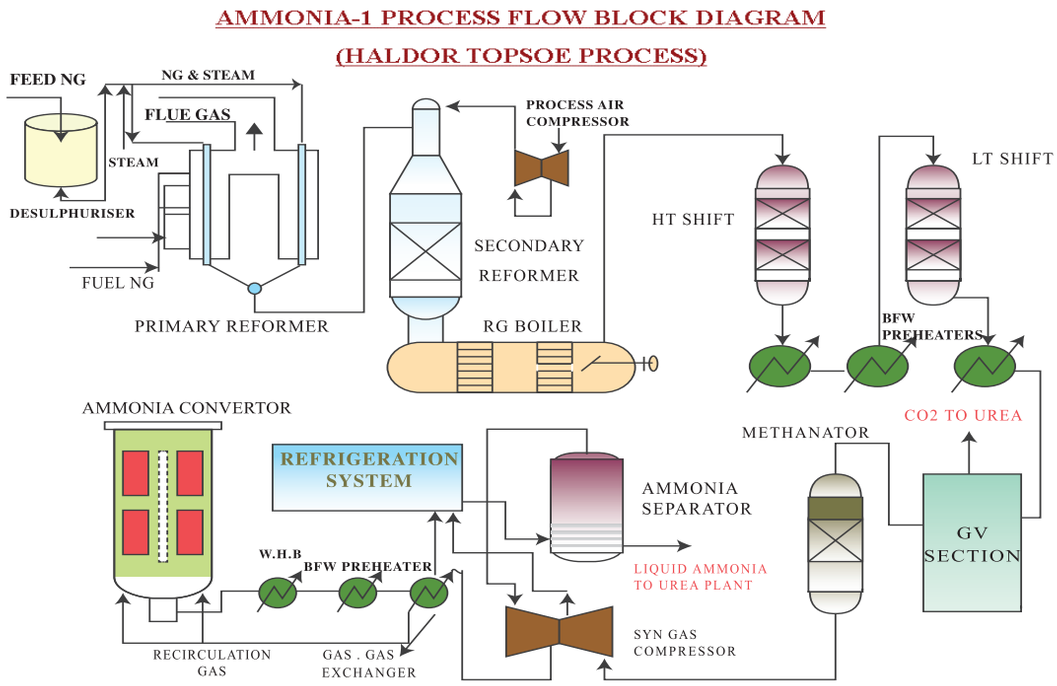


Figure 17.1 Typical Steam Reforming Process for Production of Ammonia

Feedstock Desulphurization

Hydrodesulfurization (HDS) is a catalytic process used to remove sulphur (S) from natural gas or naphtha feedstock. The important reason for removing sulfur is that sulfur, even in extremely low concentrations, poisons the downstream catalysts in steam reforming process.

The hydrodesulfurization unit consists of a fixed-bed reactor operating at 300 to 400 °C, typically in the presence of a catalyst consisting of alumina impregnated with a combination of nickel and molybdenum (NiMo) or cobalt and molybdenum (CoMo) catalyst, for specific feed stocks.
Sulphur compounds are hydrogenated to H_2S

$$R\text{-}SH + H_2 \longrightarrow H_2S + RH$$

The H_2S is then absorbed on pelletised zinc oxide

$$H_2S + ZnO \longrightarrow ZnS + H_2O$$

In this way, the sulphur is removed to less than 0.1ppm 'S' in the gas feed.

Primary reformer (Tubular reformer)

The purpose of the primary reforming is to convert bulk of the hydrocarbons feed to H_2 and CO. The overall reaction is endothermic and requires a large amount of external heat, which is provided by burning NG/naphtha fuel in the radiant chamber in which catalyst tubes are installed. The tubes are of nickel alloy, centrifugally cast and packed with catalyst. The catalyst is nickel based with promoters

such as potassium and supported on calcium aluminate base. Temperature of the tubes wall is maintained around 900 °C. There are many designs of reformer furnaces. All these aim at efficient heat transfer. The fuel burners may be placed at top, side or bottom of the furnace.

The feed gas from the desulphuriser is mixed with process steam so as to maintain a steam to carbon molar ratio (S/C ratio), at around 3.0 –3.5. Following endothermic reforming reaction takes place inside the tubes filled with catalyst.

$CH_4 + H_2O$ <---> $CO + 3H_2$ $\Delta H^0_{298} = 206$ kJ/mol

$CO + H_2O$ <---> $CO_2 + H_2$ $\Delta H^0_{298} = -41$ kJ/mol

The flue-gas leaving the radiant box has temperatures in excess of 900 °C, after supplying the necessary high level heat to the reforming process. The heat content (waste heat) of the flue-gas is used in the reformer convection section, for heating of various process streams. Fuel energy requirement in the conventional reforming process is 40-50% of the process feed-gas energy. The flue-gas leaving the convection section at 120-200 °C is one of the main sources of emissions from the plant. These emissions are mainly CO_2, NO_x, with small amounts of $SO_2$and CO.

Secondary reformer

About 10-12% of the hydrocarbon feed remains unconverted in the primary reformer because of the chemical equilibria at the actual operating conditions. In the conventional reforming process the degree of primary reforming is adjusted so that the air supplied to the secondary reformer meets both the heat balance and the stoichiometric synthesis gas requirement.

The process air is compressed to the reforming pressure and heated further in the primary reformer convection section to around 600 °C. The process gas from primary reformer is mixed with the air in a burner and then passed over a nickel-containing secondary reformer catalyst. The reformer outlet temperature is around 1,000 °C, and up to 99% of the hydrocarbon feed (to the primary reformer) is converted, giving a residual methane content of 0.2-0.3% (dry gas base) in the process gas leaving the secondary reformer. The process gas is cooled to 350-400 °C in a waste heat steam boiler or boiler/ superheater downstream from the secondary reformer.

The catalyst used for secondary reforming is similar to that of primary reforming, except it has lower nickel content (4-8 wt%) and is designed to withstand the high temperature. Rings and multi-hole pellets are the usual shapes. Either a refractory alumina or magnesium aluminate carrier is used. The catalyst is rugged with high crush strength and high temperature resistance.

Alumina balls are placed in several layers on top of the catalyst bed to prevent impinging hot gases from damaging the catalyst. Also they protect the catalyst from prolonged high temperature damage.

CO Shift Conversion – High Temperature (HTS)

The process gas from the secondary reformer contains 12-15% **CO** (dry gas base) and most of the **CO** is converted to CO_2 in the shift section according to the reaction:

$$CO + H_2O \text{ <---> } CO_2 + H_2 \qquad \Delta H^{O}_{298} = -41 \text{ kJ/mol}$$

The reaction is carried out in two stages due to equilibrium considerations. In the first stage, gas is passed through a bed of iron oxide/chromium oxide catalyst at around 400 °C, where the **CO** content is reduced to about 3% (dry gas base), limited by the shift equilibrium at the actual operating temperature. This step is called high temperature (HT) reaction. Therefore, higher CO slip from this section has penalty for energy consumption. The catalyst is an iron oxide (magnetite) catalyst structurally promoted by chromium oxide.

CO Shift Conversion – Low Temperature (LTS)

The gas from the HT Shift converter is cooled and passed through the Low Temperature Shift (LTS) converter.

This LTS converter is filled with a copper oxide/zinc oxide-based catalyst and operates at about 200-220 °C. The **residual CO** content in the converted gas is about 0.2-0.4% (dry gas base). A low residual CO content is important for the efficiency of the process. Oxides of carbon are poison to synthesis catalyst. Therefore, the residual CO and CO_2 are converted back to methane which is inert to the synthesis catalyst. The hydrogen consumed in the methanation reaction reduces the ammonia production to that extent.

LTS catalysts are high-copper catalysts based on oxides of copper, zinc and aluminum. The copper oxide is very finely dispersed resulting in high activity. Copper/zinc/aluminium LTS catalysts are seriously poisoned even by trace amounts of chlorides. Some suppliers also recommend LT catalyst guard to protect the catalyst against poisoning.

CO_2 Removal Section

After leaving the low temperature shift conversion section, the process gas contains apart from synthesis gas, around 18% **CO_2**. Other constituents i.e. CO, CH_4, inerts are present in very small quantities. The process gas stream also contains excess process steam. Separation of CO_2 from reformed gases is carried out by selective absorption of CO_2 in a solution and subsequently releasing it by depressurization and heating of the solution.

The process gas is cooled from 220°C to 50°C, in heat recovery heat exchangers and passed through CO_2 absorber operating at pressure maintained at around 30 kg/cm^2. The solution contains an activator to enhance CO_2 absorption, corrosion inhibitor and is free from suspended matter. The process gas from CO_2 absorber contains residual CO_2 around 300–2000 ppm along with other residual impurities, which are removed in the downstream gas purification section.

The rich solution leaving the absorber bottom is loaded with CO_2 and is depressurized through a pressure control valve / hydraulic turbine. It is then heated through a number of heat recovery heat exchangers and then sent to the regenerator, operating at slightly higher than atmospheric pressure. CO_2 gets released, which is then cooled in heat recovery exchangers and sent to urea plant.

Process condensate is stripped of all these chemicals by means of steam and the stripped effluent water is recycled in the process.

In the CO_2 absorption solution, activators viz DEA, glycine etc; are added to enhance CO_2 absorption. Further, the CO_2 rich solution at boiling temperature is highly corrosive. Therefore, carbon steel equipment are provided with passivation layers (oxidation layers) by adding corrosion inhibitor i.e. titanium oxide. Side stream filters are provided to remove suspended matter from the solution.

Methanation

The small amounts of CO and CO_2, remaining in the synthesis gas, are poisonous for the ammonia synthesis catalyst and must be removed by conversion to CH_4 in the methanator.

The reactions take place at around 300 °C in a reactor filled with a nickel containing catalyst. Following reactions take place.

$CO + 3H_2$ **----->** $CH_4 + H_2O$

$CO_2 + 4H_2$ -----> $CH_4 + 2H_2O$

Residual CO and CO_2 are converted to methane, which acts as inert to the downstream ammonia synthesis catalyst. The catalyst is nickel based, dispersed on silica, calcia and magnesia. Nickel content is about 20-34%. It comes in various physical shapes i.e. spheres, tablets, extrusions, rings etc.

Gas purifier

An alternative to methanation for removal of CO and CO_2 from process gas is cryogenic purification. Cryogenic purification also removes residual methane and argon in the synthesis gas. The process involves washing of gas with liquid nitrogen. Cryogenic system operates at temperature of minus 170°C to minus 200°C. The off-gas is used as fuel in reformer furnace.

Ammonia synthesis

The purified process gas is called “make up synthesis gas”. It contains hydrogen and nitrogen in the mole ratio of 3:1. It also contains some inerts i.e. methane and argon. The gas is compressed in synthesis gas compressor to synthesis loop pressure of 120 – 200 Kg/cm^2. The gas entering the ammonia converter consists mainly of unconverted gas recirculated in the loop along with “Make up” gas . The gas entering the converter contains H_2 and N_2 in the stoichiometric ratio of 3:1 plus 10-14% inerts and about 2% ammonia. The inerts consist mainly of methane, argon and sometimes helium. The synthesis of ammonia takes place by following reversible reaction

$$N_{2(g)} + 3H_{2(g)} \rightleftharpoons 2NH_{3(g)} \quad \Delta H = -92 kJmol^{-1}$$

Operating temperature is 400-450 ^oC and the catalytic reaction is exothermic. The metallic iron catalyst is primarily made from magnetite, Fe_3O_4, that has been promoted using alkali in the form of

potash and metals such as aluminium, calcium and magnesium. The catalyst is susceptible to permanent poisoning by sulphur, arsenic, phosphorus, chlorine and heavy hydrocarbons; while oxygen bearing compounds will cause temporary poisoning.

Ammonia synthesis is affected by pressure, temperature, inlet gas composition, space velocity and catalyst particle size etc. LeChatelier's principle determines the reaction rate and equilibrium ammonia concentration. Thus, raising the pressure increases the equilibrium percentage of ammonia and accelerates the reaction rate. Reaction being exothermic, higher temperatures increase reaction rate but equilibrium percentage of ammonia decreases. Reaction rate is also affected by other factors like ammonia concentration in the incoming feed gas, presence of inerts, make up gas composition, catalyst activity and its particles size, etc.

The gas leaving the converter contains 12-18% ammonia. By continual recycling of the un-reacted nitrogen and hydrogen, the overall conversion is about 98% of make-up gas.

The heat contained in the gas leaving ammonia converter at around 400 ^{o}C is utilized in a series of successive heat exchangers. The higher level of heat is recovered by raising steam at 105 kg/cm^2 and then heating boiler feed water. It is then used to heat incoming synthesis gas feed to converter. After effective heat recovery, the gas is cooled with cooling water and further cooled by refrigeration, to condense most of ammonia as liquid. Liquid ammonia is separated and the unconverted gas is recycled back to the converter through recycle compression stage of synthesis gas compressor. The ammonia vapours generated, are compressed and liquefied in the refrigeration section.

The inerts (methane and argon) do not dissolve in the liquid ammonia. Their accumulation in the synthesis loop retards reaction rate. Thus the major portion of these inert gases is removed by taking out a purge stream from the loop. The purge gas has a typical composition of 60% H_2, 20% N_2, 13% CH_4, 4% Ar and 2% NH_3.

In the older plants, purge gas was used as fuel in the reformer furnace. Over the period, purge gas is re-processes to recover hydrogen by one of the processes viz. (i) cryogenic process (ii) Membrane separation (iii) Pressure Swing Adsorption (PSA).

Ammonia Storage

In the older plants of lower capacity, ammonia is stored in pressurized systems such as bullets and in Horton spheres.

With the large scale industrial production of ammonia, it has become common to store ammonia at atmospheric pressures at -33°C. Atmospheric ammonia storage requires lesser capital per unit volume and is safer than sphere storage using pressures higher than atmospheric.

17.2.2 Urea

Urea is manufactured by reacting liquid ammonia with CO_2 gas at high pressure (160 – 250 ata) and temperature (190 ^{O}C). The reaction is non-catalytic. Distinct process steps for manufacturing urea are described below (Figure 17.2).

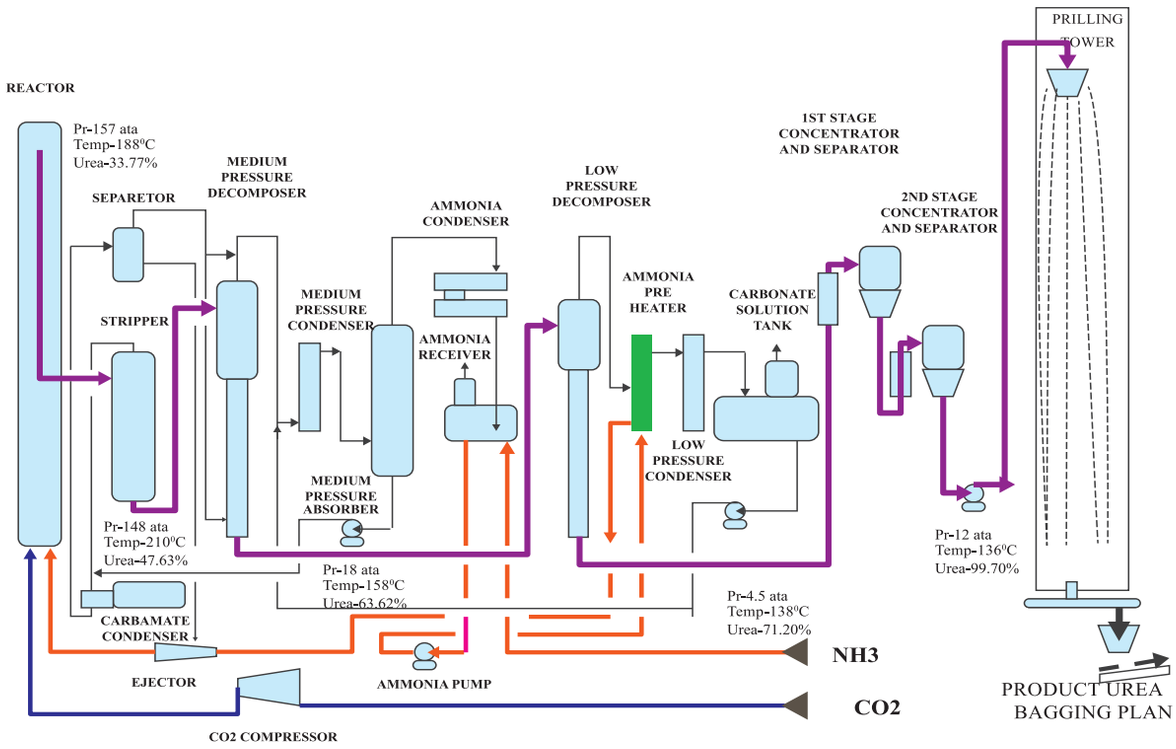


Figure 17.2 Typical Urea Synthesis Process Flow Digaram

Urea synthesis

Urea is produced by synthesis from liquid ammonia and gaseous carbon dioxide. These two react to form ammonium carbamate, a portion of which dehydrates to urea and water. The reactions are as follows:

$$2NH_3 + CO_2 \longrightarrow NH_2COONH_4$$

$$NH_2COONH_4 \longrightarrow NH_2CONH_2 + H_2O$$

Hence, overall reaction is,

$$2\ NH_3 + CO_2 \longrightarrow NH_2CONH_2 + H_2O$$

In synthesis condition (T=190ºC, P = 160 kg/cm²), first reaction occurs rapidly and is completed, second reaction occurs slowly and determines reactor volume. The Mole ratio of ammonia to carbon dioxide is 3.6 to 1. A small amount of air is added at compressor's suction for passivation of Stainless Steel surfaces.

Recovery of un-converted reactants

In the urea reactor, the reaction is complete by 60-70% and due to recycling of unconverted reactants, urea concentration in products coming out of reactor is 30-35%.

The unconverted reactants are separated out from urea solution by reducing pressure and supplying external heat by steam, usually accomplished in three stages viz synthesis pressure, 12-18 ata and 4.5 ata. The vapours so generated are condensed to obtain carbamate solution and the process is highly exothermic. Utilisation of heat of condensation of carbamate vapours is the genesis of improving thermal efficiency of urea process. In most of the cases, steam is generated at low pressure (4.5ata).

Urea purification and low pressure recovery

Residual carbamate is decomposed in Medium Pressure (12-18 ata) & Low Pressure (4.5 ata) decomposers by heating with steam / hot condensate. The carbamate vapours leaving the top of decomposer, are sent to the condenser where they are absorbed in recycled aqueous carbonate solution. The absorption and condensation heat is removed by cooling water. The inert gases which essentially contain ammonia vapour now directly into low pressure falling film absorber, where the ammonia is absorbed by a counter current water flow.

Urea concentration section

Solution leaving low pressure decomposer bottom with about 72% urea is concentrated in two vacuum stages operating at 0.3ata and 0.03ata. Urea solution is concentrated upto 99.5 %.

Urea prilling

The final finishing of urea is carried out in prilling tower which are tall concrete structures. These can be natural or forced/induced draught. Melted urea leaving second vacuum separator is sent to Prilling bucket at top. In natural draught droplets falling get cooled and solidify by the ambient air entering from the bottom of the prilling tower. Some of the old generation plants have prilling towers with height lower than natural drafts and operate with provision of induced/ forced draught fans for cooling of the prills.

The solid prills falling to the bottom of prilling tower are sent to product handling section / storage through belt conveyor.

17.2.3 Other Nitrogenous Fertilisers

Other nitrogenous products i.e. ammonium sulphate (AS), calcium ammonium nitrate (CAN), ammonium chloride (ACl) are mainly produced as by-products of other chemical plants. Currently, there is only one operating plant producing CAN. Ammonium sulphate is obtained as by product of caprolactum plants and in small quantities from steel plants.

17.3 Energy flow

The fertiliser making process is highly energy intensive accounting to nearly 60-70% of the production cost. This provides ample opportunities for reducing energy consumption.

17.3.1 Thermal Energy

Ammonia is produced from a gaseous mixture of nitrogen and hydrogen. Nitrogen is derived from atmospheric air and hydrogen is obtained mainly from fossil fuels. In exceptional cases where electricity is cheaper, hydrogen can also be obtained by water (electrolysis).

Thus, in ammonia manufacturing process, fossil fuel energy is used as "Feedstock" as well as "Fuel".

Feedstock energy

Main consumer of fossil fuel energy is primary reformer where natural gas / naphtha are consumed as feedstock for production of ammonia.

Fuel energy

Fuel energy is consumed in following systems:

(i) Reformer furnace for supplying endothermic heat of steam reforming reaction.
(ii) Captive power plant for power generation.
(iii) Boiler for raising steam.
(iii) In some plants, fuel fired furnaces are used for pre-heating of natural gas, process air or superheating steam.

Steam production

(i) Steam production in ammonia plant: Steam reforming ammonia plants have process gas at high temperature at outlet of (a) secondary reformer at 950-1000 ^{o}C (b) high temperature shift converter at 400–450 ^{o}C (c) synthesis converter at 400-450 ^{o}C. This heat is recovered by producing steam at 105-110 kg/cm^2 and also superheating the same up to about 505 ^{o}C.

(ii) Gas turbine / Process air compressor / Heat recovery unit: In some plants installed since 1990s, gas turbine is provided to drive process air compressor and steam is produced at 105 kg/cm^2 from exhaust of the gas turbine.

(**iii) Auxiliary / Service boiler**: A standalone auxiliary boiler is provided to meet additional steam requirement, specifically during startup of the plant.

Steam utilization

Steam at high pressure and temperature (105 kg/cm^{2,} 505 ^{0}C), is utilized to drive turbine for synthesis gas compressor. Steam extraction at medium pressure (40 kg/cm^2) is used mainly as reformer feed and for driving process air compressor, refrigeration compressor, CO_2 compressor. It is also used to drive high rating pumps for boiler feed water (BFW), cooling water (CW) etc.

From bigger turbines, steam is also extracted at different pressure levels to drive smaller turbines as well as heating the process fluids.

In modern design of ammonia plant flow sheet, emphasis is laid on optimization of steam production and consumption not only for ammonia plant battery limit, but for the entire fertiliser complex.

17.3.2 Electrical Energy

Almost all the Indian ammonia plants are having captive power plants so as to avoid disruptions in plant operation due to power interruptions which are more likely from grid power. However, these plants are also connected with power grid to meet contingencies and draw minimum power to be used in township and other non-plant use.

There are three models of generating captive power.

(i) Where natural gas is available in sufficient quantity, gas turbine is coupled with power generator and steam is produced by heat recovery from exhaust gases with supplementary firing. Rating of turbo generator is up to 25 MW at 11 kV under ISO conditions. Normally two sets are provided. In India, almost all the gas based ammonia plants installed since 1980s are having gas turbo-generators / Heat Recovery Units (HRU).

(ii) Where supply of natural gas is restricted or not available at all, steam is generated in boilers by firing alternative fuels like furnace oil, coal etc. Steam turbo generators are provided.

(iii) Plants are also having diesel generating sets of capacity around 5-7 KVA to be used as "Emergency power", to run essential services, in case of tripping of main power supply.

17.4 Material and Energy Balance

17.4.1 A typical material and energy flow diagram of an ammonia/urea fertiliser complex is illustrated below in Figure 17.3

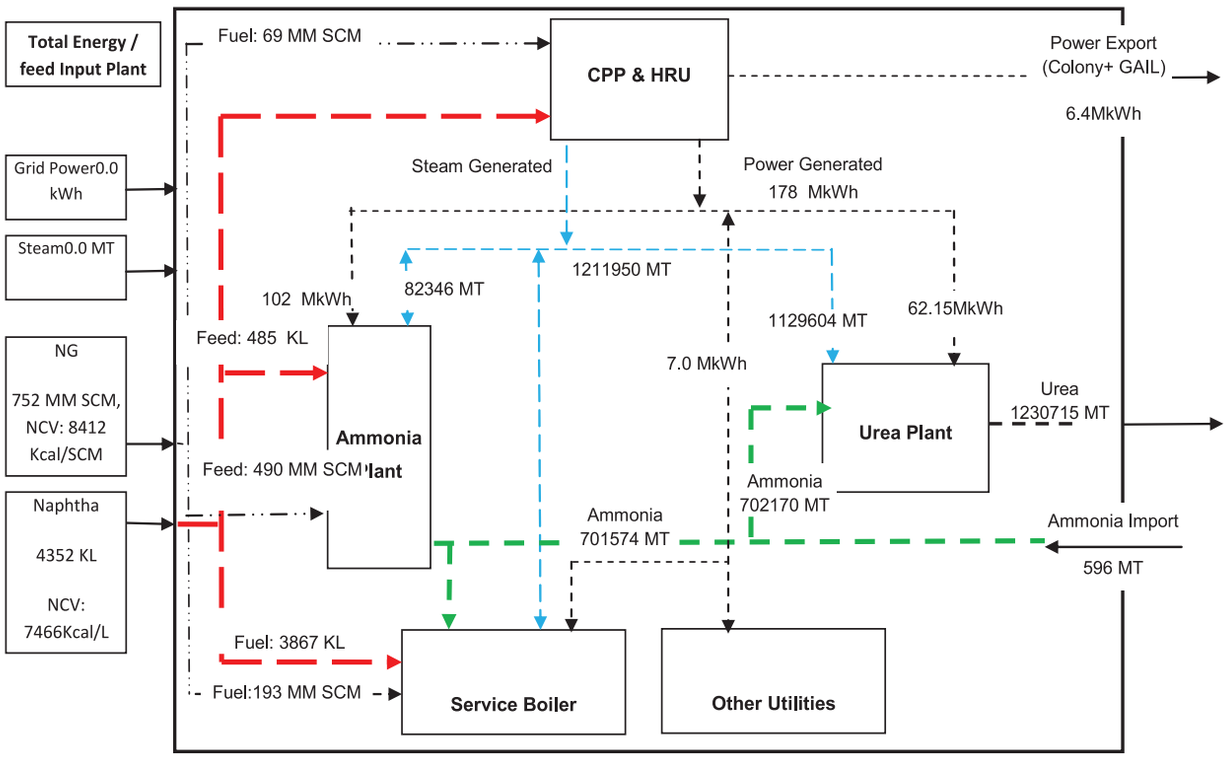


Figure 17.3 Material & Energy balance flow diagram of ammonia/urea fertiliser complex

An Illustration:

Input Data

(a) Overall plant battery limit

Inputs

Natural gas (NG)	752 MM SCM
NCV of NG	8412 Kcal/SCM
Naphtha	4352 KL
NCV of Naphtha	7466 kCal/L(10575 kCal/kg)
Density of Naphtha	0.71 g/cc
Grid power (MWh)	0
Steam (MT)	0
Ammonia	596 Tonnes

Output

Urea product	1230715 Tonnes
Power export	6.4MkWh

(b) CPP/HRU

Input

Natural gas	69 MM SCM

Output

Power to plant	178 MkWh
Power export	6 MkWh
Heat rate	511 kcal/kWh
Steam	1211950 Tonnes
Heat content	743 kCal/Kg

(c) Ammonia plant

Input

Natural gas as feed	490 MM SCM
Natural gas as fuel	130 MM SCM
Naphtha as feed	485 KL
Naphtha as fuel	1626 KL
Steam	82346 Tonnes
Power	102 MkWh

Output

Ammonia product	701574 Tonnes

(d) Urea plant

Input

Ammonia	702170 Tonnes
Steam	1129604 Tonnes
Power	62 MkWh

Output

Urea product	1230715 Tonnes

(e) Service boiler / utilities

Input

Naphtha fuel	3867 KL
Natural gas	193 MM SCM

Calculate specific energy consumption for ammonia as Gcal/Tonnes ammonia
Calculate specific energy consumption for urea as Gcal/Tonnes urea

Solution:

(a) Material balance of natural gas

Description	Quantity (MM SCM)	NCV (kcal/SCM)	Heat content (Gcal) 2 x 3
1	2	3	4
Total input			
	752	8412	6325824
Distribution			
CPP/HRU	69	8412	580428
Reformer feed	490	8412	4121880
Reformer fuel	193	8412	1623516

(b) Material balance of naphtha

Description	Quantity (KL)	NCV (kcal/ L)	Heat content (Gcal) 2 x 3
1	2	3	4
Total input			
	4352	7466	32492
Distribution			
Reformer feed	485	7466	3621
Service boiler	3867	7466	28871

(c) Energy balance in Ammonia Plant

Item	Unit	Quantity	NCV/ Heat value (kcal/Sm^3)	Heat content Gcal	Specific energy consumption Gcal/Tonne
Ammonia production	Tonnes	700978			
NG feed	MM SCM	490	8412	4121880	5.880
NG fuel	MM SCM	130	8412	1093560	1.560
Naphtha feed	Tonnes	342	10575	3617	0.005
Naphtha fuel	Tonnes	1154	10575	12204	0.018
Steam	Tonnes	82346	743	61183	0.088
Energy	MkWh	101917	511	52080	0.074
Total feed energy				4125497	5.885
Total fuel energy				1105764	1.578
Power + steam				113263	0.162
Total SEC				5344523	7.625
Non plant energy					0.066
SEC including non plant energy					7.691

(d) Energy balance in Urea Plant

Item	Unit	Quantity	NCV/ Heat value	Heat content Gcal	Specific energy consumption Gcal/Tonne
Urea production	Tonnes	1230715			
Ammonia feed	Tonnes	701574	7.691 (Gcal/Tonne)	5395806	4.384
Steam	Tonnes	1129604	7435 kCal/Tonne	839295	0.682

Power	MkWh	62150	511 kcal/ Tonne	31759	0.026
Total energy				6266860	
Total SEC					5.092
Non plant energy					0.079
SEC including non plant energy					5.171

17.4.2 Ammonia / urea complex Steam Balance - Illustration

Brief description of steam network

In an ammonia / urea complex, steam is generated at high pressure (105 kg/cm^2, 515 ^{o}C), mainly in non-fired heaters viz. Reformed gas boiler (RG boiler) and Ammonia synthesis boiler (Synthesis boiler) where vast quantity of heat is available from highly exothermic reactions. High pressure steam is also produced in direct fired auxiliary boilers located either in ammonia plant or Off-sites. Total steam produced is fed into a common HP steam header being maintained at 105 kg/cm^2, 515 ^{o}C.

HP steam is consumed in steam turbines coupled with gas compressors. These are turbines are of extraction type, condensing type or a combination of both, depending on steam requirement in the downstream facilities. In most of ammonia plants, whole of HP steam is fed into synthesis gas compressor turbine, from where steam is extracted at medium pressure (40 kg/cm^2). A little quantity of HP steam is sent through by-pass throttling valve to keep the by-pass steam header hot and by-pass valve in floating condition.

Medium pressure (MP) steam is utilized mainly as reformer feed along with natural gas to produce reformed gas for production of ammonia. It is also used for running steam turbines of process air compressor, refrigeration compressor, cooling water pumps, boiler feed water pumps, ID/FD fans for reformer and other pumps/fans of medium duty.

In the steam condensing turbines, exhaust steam is condensed in water cooled condenser and thus, a considerable quantity of heat is lost in cooling water sink. On the other hand, in steam extraction turbines, there is no heat loss to the sink but there must exist consumers of extracted steam.

Typical steam balance diagram

A typical steam balance diagram (Figure 17.4) in an ammonia/urea complex is given below. The diagram is self- explanatory. Impact of quench water added for de-superheating of the superheated steam obtained after expansion, is ignored.

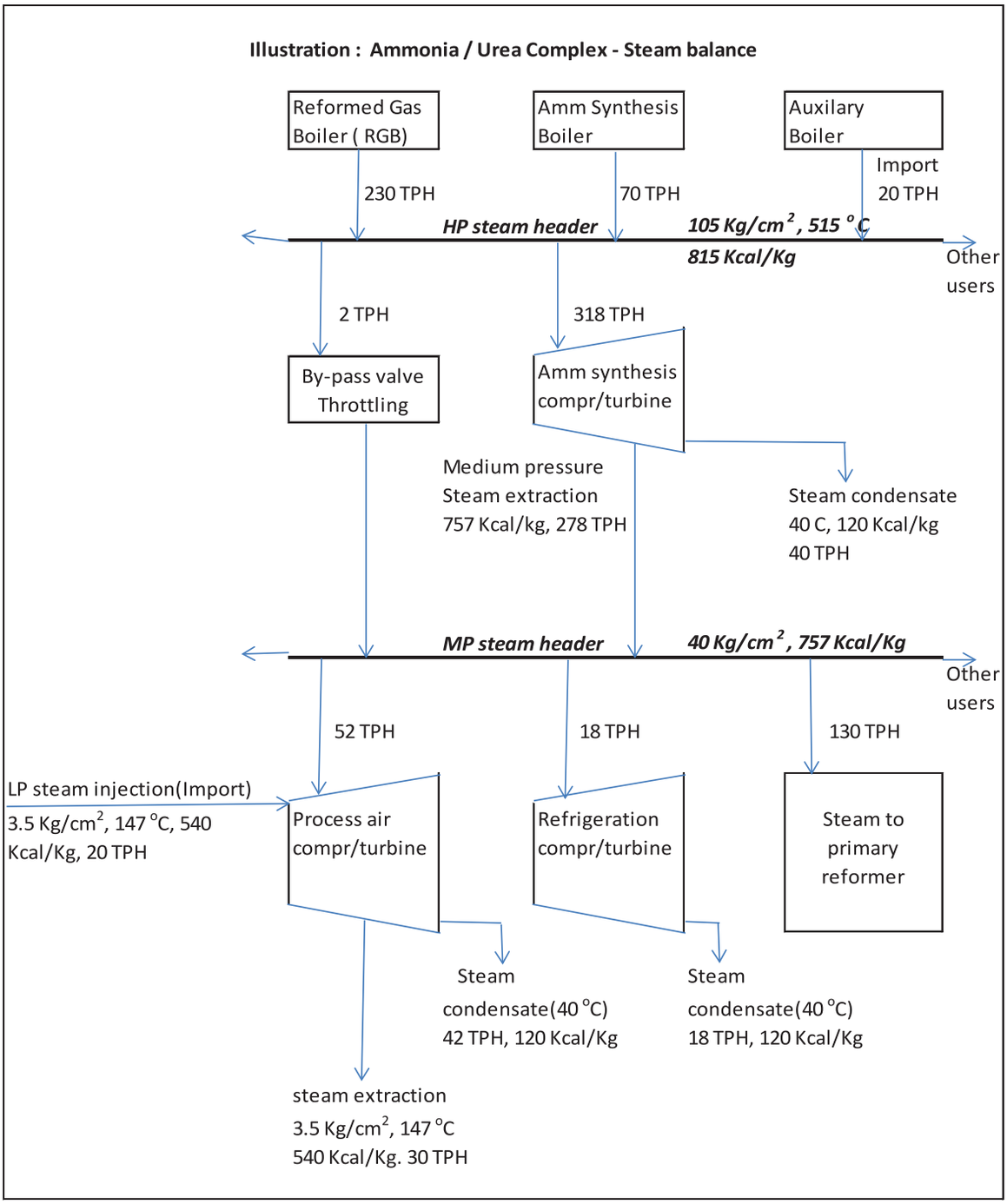


Figure 17.4 Steam balance flow diagram for ammonia/urea complex

From the steam balance diagram given above, calculate the following:

A. Energy balance in HP steam header.
B. Energy balance around ammonia synthesis turbine.
C. Energy balance in MP steam header.
D. Energy balance around process air compressor.
E. Total energy lost in heat sink of cooling water.

Solution:

A. Energy balance in HP steam header

S.No.	Item	Quantity	Heat value	Total energy
		TPH	kcal/kg	Gcal/hr
1.0	**Steam input to HP steam header**			
1.1	RG boiler	230	815	187.45
1.2	Synthesis boiler	70	815	57.05
1.3	Auxiliary boiler	20	815	16.30
1.4	**Sub-total**	**320**	**815**	**260.80**
2.0	**Steam output from HP steam header**			
2.1	Inlet to synthesis compressor/turbine	318	815	259.17
2.2	By-pass throttling	2	815	1.63
2.3	Others (if any)	0	815	0.00
2.4	**Sub-total**	**320**	**815**	**260.80**

B. Energy balance around ammonia synthesis turbine

S.No.	Item	Quantity	Heat value	Total energy
		TPH	kcal/Kg	Gcal/hr
1.0	**Steam input to Synthesis turbine/compressor**	318	815	259.17
2.0	**Steam output from synthesis turbine/ compressor**			
2.1	MP steam extraction (40 kg/cm^2)	278	757	210.45
2.2	Condensing steam	40	120	4.80
2.3	**Sub-total**	**318**	**----**	**215.25**
3.0	**Thermal energy converted to mechanical energy**	**----**	**----**	**43.92**

C. Energy balance in MP steam header

S.No.	Item	Quantity	Heat value	Total energy
		TPH	kcal/kg	Gcal/hr
1.0	**Steam input to MP steam header**			
1.1	Extraction from synthesis turbine/compressor	278	757	210.45
1.2	By-pass valve throttling	2	757	1.52
1.3	**Sub-total**	**280**		**211.97**
2.0	**Steam output from MP steam header**			

2.1	Steam input to process air compressor	52	757	39.37
2.2	Steam input to Refrigeration compressor	18	757	13.63
2.3	Steam feed to reformer	130	757	98.41
2.4	Other users	80	757	60.56
2.4	**Sub-total**	**280**	757	**211.97**

D. Energy balance around process air turbine/compressor

S.No.	Item	Quantity	Heat value	Total energy
		TPH	kcal/kg	Gcal/hr
1.0	**Steam input to Process Air turbine/compressor**			
1.1	MP steam	52	757	39.36
1.2	LP steam injection	20	540	10.80
1.3	**Sub-total**	**----**	**----**	**40.16**
2.0	**Steam output from Process Air turbine/ compressor**			
2.1	LP steam extraction (40 kg/cm^2)	30	540	16.20
2.2	Condensing steam	42	120	5.04
2.3	**Sub-total**	**----**	**----**	**21.24**
3.0	**Thermal energy converted to mechanical energy**	**---**	**---**	**18.92**

E. Total energy lost in heat sink of cooling water

S.No.	Item	Quantity	Heat value	Total energy
		TPH	kcal/kg	Gcal/hr
1.0	Synthesis turbine condensing stage	40	120	4.80
2.0	Process Air turbine condensing stage	18	120	2.16
3.0	**Total**	**58**		**6.96**

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